
Air Conditioning

Professional Engineer Examination Review – Mechanical

Center for Continuing Engineering Education (C2E2)
web.odu.edu/engr/ccee

Definitions:

Dry Bulb Temperature - The temperature of a mixture of gasses indicated by an accurate thermometer.

Wet Bulb Temperature - Temperature at which liquid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature

Adiabatic Process - No net heat is added to or taken from a substance or system undergoing the process.

Dew Point Temperature - Temperature at which condensation will just begin when the moist air mixture under consideration is cooled at constant pressure

Humidity Ratio or Specific Humidity - The mass of water vapor per unit mass of dry air in a vapor-air mixture $\text{GRAINS H}_2\text{O/LB D.A.}$ 7000 grains = 1 pound

Relative Humidity - The ratio of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature.

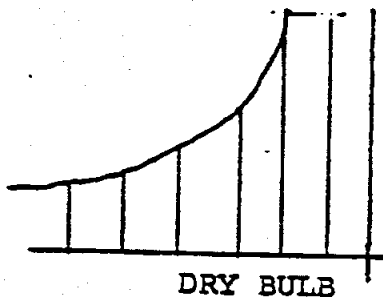
Sensible Heat Load - Heat added or removed that changes only the temperature of the air vapor mixture.

Latent Heat Load - Heat required to evaporate water vapor into or remove water vapor from the air vapor mixture.

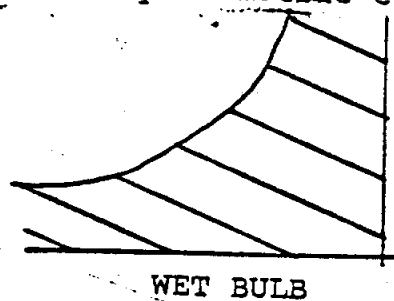
Sensible Heat Ratio - Ratio of the sensible heat load to the total heat load of the space considered.

Standard Air - Air with a density of 0.075 lb. per cubic foot. This is substantially equivalent to dry air at 70°F and 29.92 in.Hg. barometer.

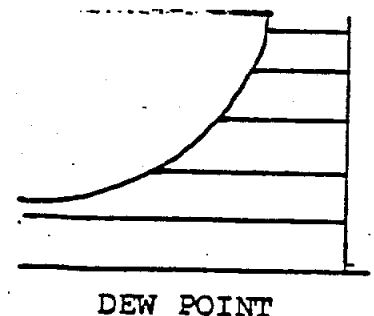
Lines on Psychrometric Chart



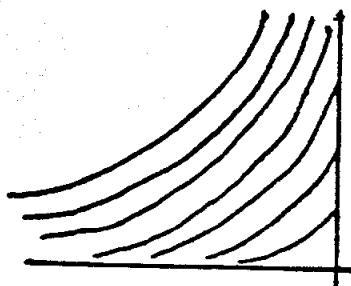
DRY BULB



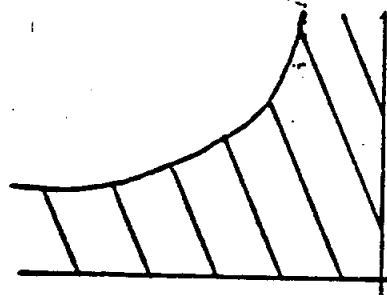
WET BULB



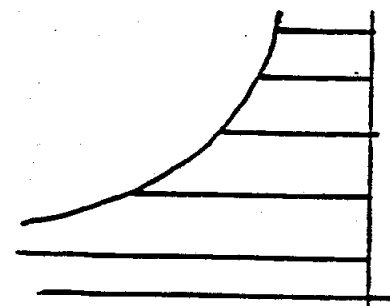
DEW POINT



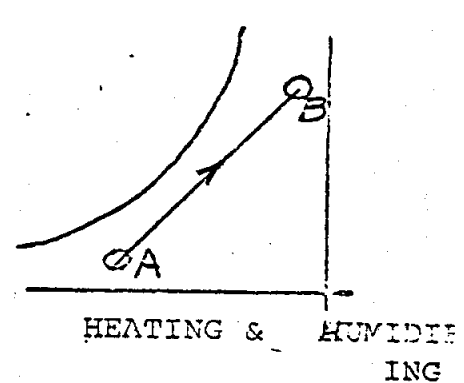
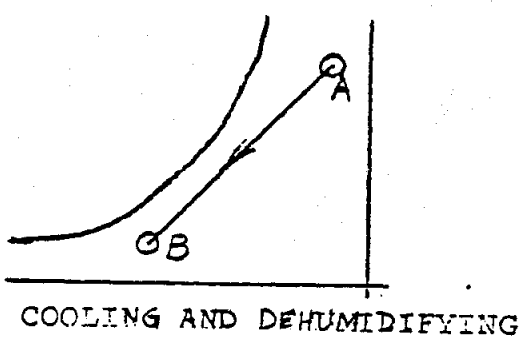
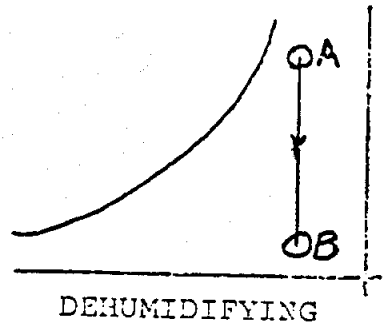
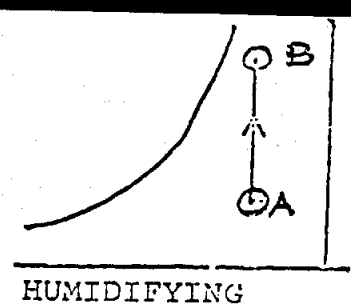
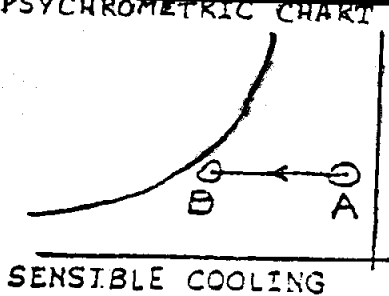
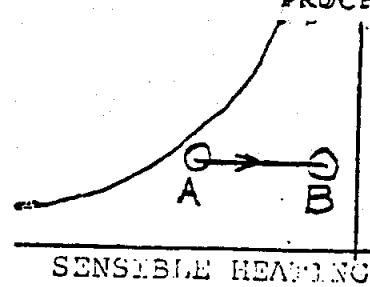
RELATIVE HUMIDITY



SPECIFIC VOLUME



HUMIDITY RATIO

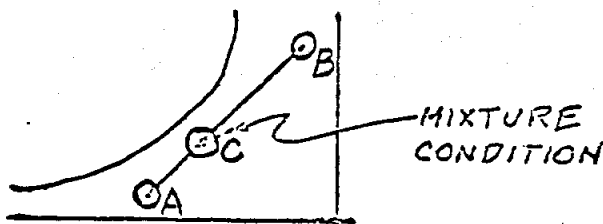


AIR MIXTURES

The resulting condition of a mixture of two air supplies at different conditions can be found on the psychrometric chart. First plot the two conditions on the chart and draw a line between them. The resulting condition will be on this line at the dry bulb calculated below.

$$\frac{\text{CFM "A" air} \times \text{D.B. "A" air}}{\text{CFM Total}} + \frac{\text{CFM "B" air} \times \text{D.B. "B" air}}{\text{CFM Total}} =$$

Sum of numbers = D.B. Mixed Air =



If the three conditions and the total CFM are given the CFM of each component can be found by a ratio of the dry bulb temperatures

$$\text{CFM "A"} = \text{Total CFM} \times \frac{t_B - t_C}{t_B - t_A}$$

$$\text{CFM "B"} = \text{Total CFM} \times \frac{t_C - t_A}{t_B - t_A}$$

U.S. (I.P.) UNITS
AIR CONDITIONING
"MAGIC" NUMBERS

$$Q_{\text{SENS}} = 1.08 \times \text{CFM} \times \Delta T_{\text{SENS}} (^{\circ}\text{F}) \quad \text{BTU/HR}$$

$$Q_{\text{LAT.}} = .68 \times \text{CFM} \times \Delta W \left(\frac{\text{GR}}{\text{LB}_{\text{DA}}} \right) \quad \text{BTU/HR}$$

$$Q_{\text{TOTAL}} = 4.5 \times \text{CFM} \times \Delta h \quad \text{BTU/HR}$$

$$1.08 = 4.5 \times .24$$

$$\frac{\text{FT}^3}{\text{MIN}} \times 60 \frac{\text{MIN}}{\text{HR}} \times .075 \frac{\text{LB}}{\text{FT}^3} = 4.5 \times \text{CFM} \quad \frac{\text{LB}}{\text{HR}}$$

$$.68 = 1060 \frac{\text{BTU}}{\text{LB}_{\text{H}_2\text{O}}} \times \frac{\text{FT}^3}{\text{MIN}} \times \frac{\text{GR}_{\text{H}_2\text{O}}}{\text{LB}_{\text{DA}}} \times 60 \frac{\text{MIN}}{\text{HR}} \times .075 \frac{\text{LB}}{\text{FT}^3} \times \frac{1}{7000} \quad \frac{\text{LB}}{\text{GR}}$$

When conditioned air enters a room it mixes with the room air and comes into equilibrium with the heat load of the room. The cool supply air is warmed by the various sources of heat in the room. The resulting equilibrium temperature is the design condition of the room.

The sensible heat ratio $\left(\frac{\text{sensible heat gain}}{\text{total heat gain}} \right)$ is an index of the type of load. If the ratio is high it is a dry type of load (telephone equipment room). If it is low, it is a wet type of load (laundry). To maintain conditions in a room the air supply must be on the sensible heat ratio line.

To draw the sensible heat ratio line thru any point on the chart, line up the reference point with the calculated sensible heat ratio on the scale to the right of the chart. This line has the correct slope. Now, a line parallel to this one and thru the desired condition will give the true sensible heat ratio line. To maintain the desired conditions the supply air must be on this line.

If the dry bulb of the supply conditions is known the CFM required can be found from the sensible heat equation

$$Q_{\text{SENSIBLE}} = 1.08 \text{ CFM} (t_2 - t_1) \quad \text{BTU/HR}$$

$$\text{or CFM} = \frac{Q_{\text{SENSIBLE}}}{1.08 (t_2 - t_1)} \quad 1.08 = 4.5 \times .24$$

The total load equation uses the change in enthalpy between two conditions.

$$Q_{\text{TOTAL}} = 4.5 \text{ CFM} (h_2 - h_1)$$

$$\text{or CFM} = \frac{Q_{\text{TOTAL}}}{4.5 (h_2 - h_1)}$$

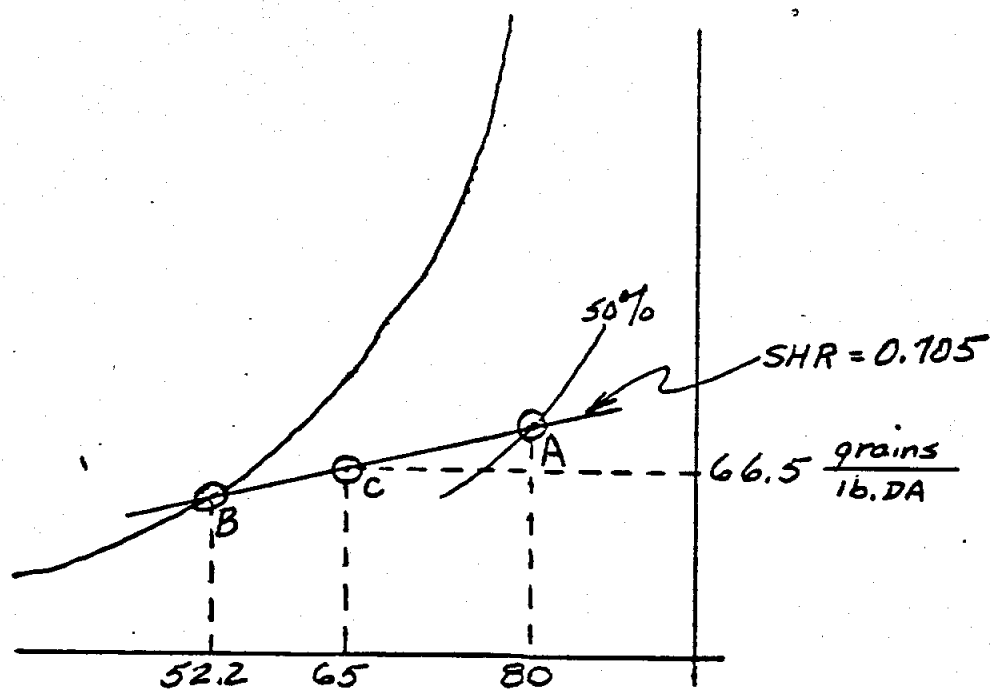
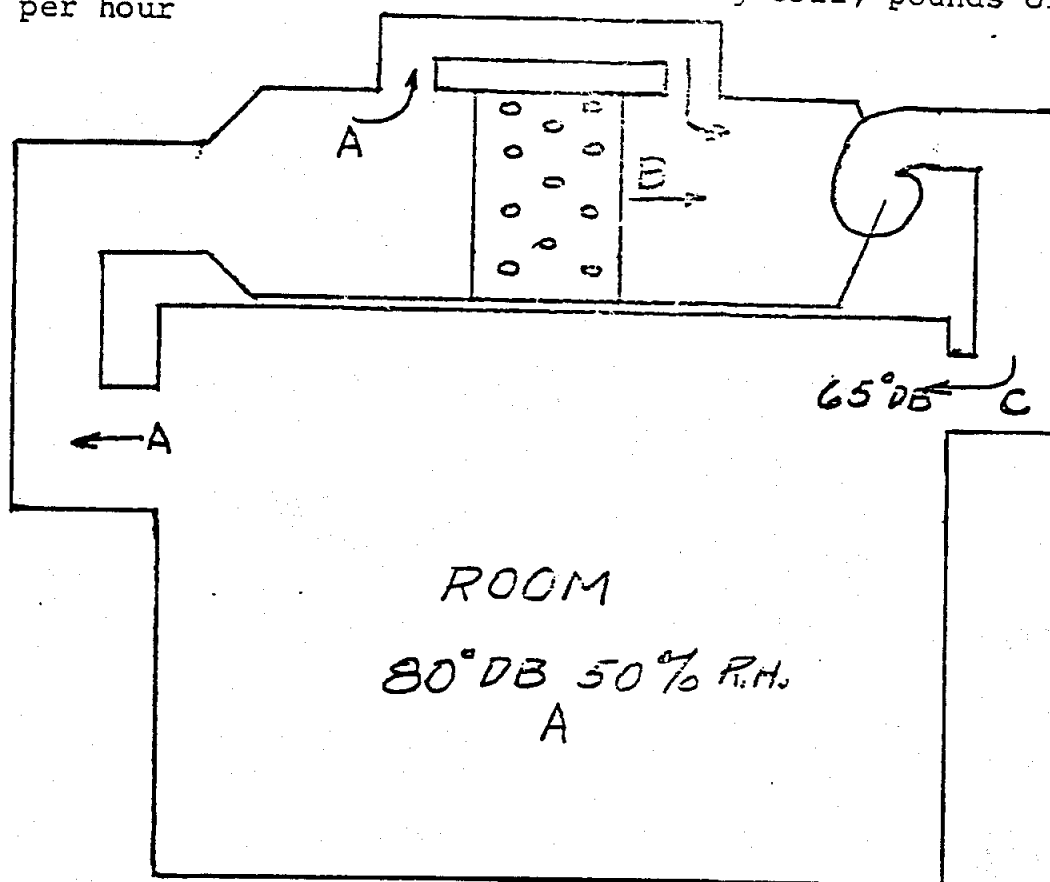
$$\frac{\text{FT}^3}{\text{MIN}} \times 60 \frac{\text{MIN}}{\text{HR}} \times .075 \frac{\text{LB}}{\text{FT}^3} =$$

$$4.5 \times \text{CFM}$$

PROBLEM:

A certain space is to be air-conditioned with supply air entering at 65 dbt. Sensible and latent heat loads are 600,000 and 250,000 Btu/hr, respectively. Air leaving the conditioned space is to be at 80 dbt and 50% relative humidity. The room air is to be recirculated through a cooling coil except that enough room air is to by pass the cooling coil to satisfy the specific humidity requirements of the supply state. With zero requirements for ventilating air, and assuming that the air which leaves the cooling coil is saturated at the coil surface temperature (apparatus dewpoint), determine:

- mass flow rate of supply air, pounds of dry air per hour
- specific humidity of supply air, grains/pound, dry air
- mass flow rate of air through the cooling coil, pounds of dry air per hour



$$\text{SHR} = \frac{600,000}{850,000} = 0.705$$

$$\text{a. CFM} = \frac{Q_s}{1.08(t_A - t_c)} = \frac{600,000}{1.08(80 - 65)} = 37,000 \text{ CFM}$$

$$4.5 \times 37,000 = 166,500 \frac{\text{lb}}{\text{hr.}}$$

b. From Psychrometric chart at pt. C

Specific Humidity = 66.5 grains per lb. Dry Air

c. From Psychrometric Chart t_{db} at pt. B = 52.2°F

$$\text{CFM} = \frac{Q_s}{1.08(t_A - t_B)} = \frac{600,000}{1.08(80 - 52.2)} = 20,000 \text{ CFM}$$

$$4.5 \times 20,000 = 90,000 \frac{\text{lb.}}{\text{hr.}}$$

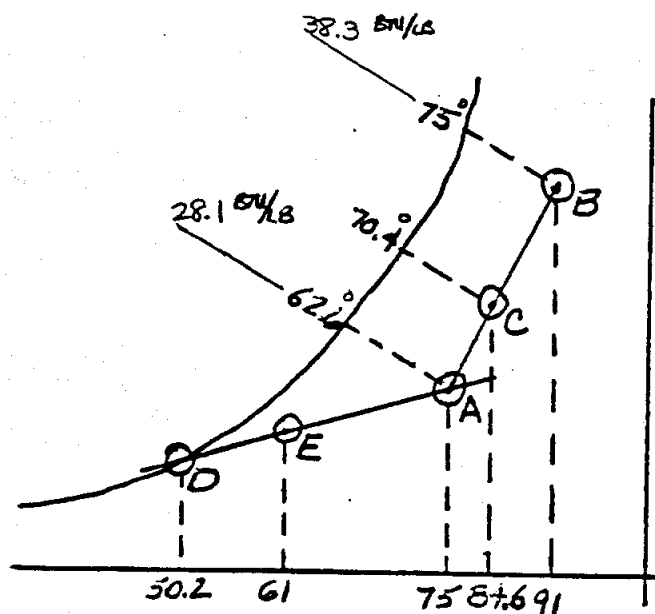
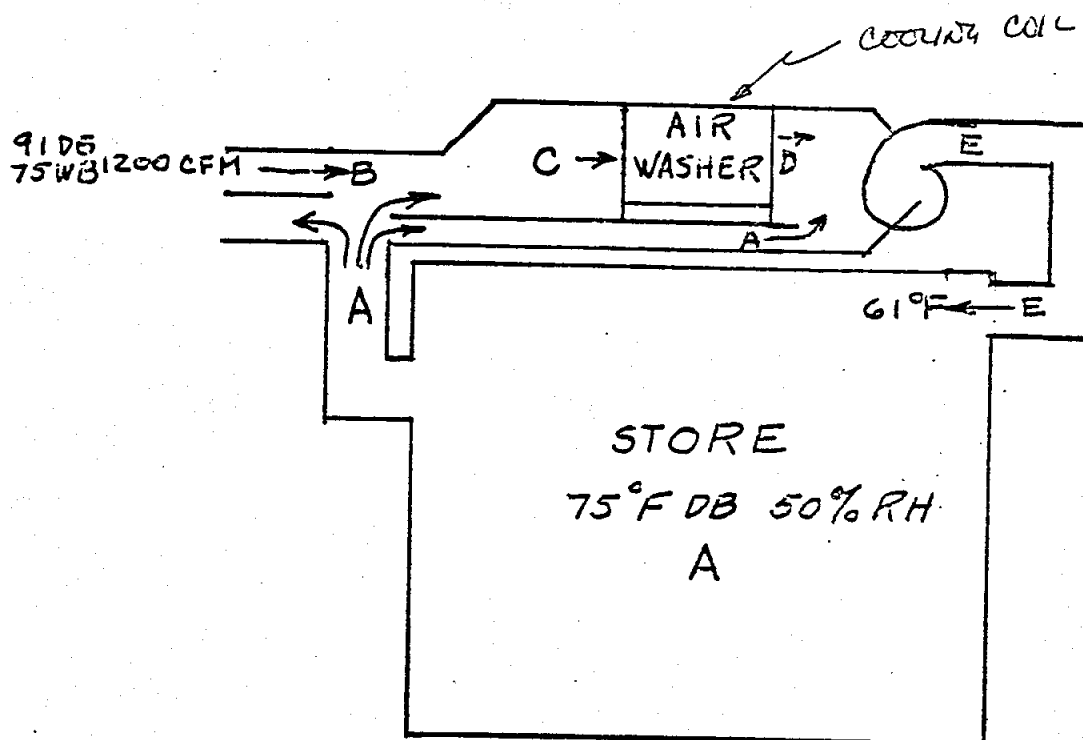
ALL HEAT ($Q_s + Q_L$) IS REMOVED IN COIL

Air by passed around coil would be:

$$37,000 - 20,000 = 17,000 \text{ CFM}$$

A small store has a total heat gain from all sources, except outdoor air, of 67,500 Btu per hr. The latent heat portion of the above is 14,400 Btu per hr. Required indoor air state is to be 75°F dry bulb and 50% relative humidity. The outdoor air state is 91°F dry bulb and 75°F wet bulb. 1200 cfm outdoor air is supplied to the system and mixed with a part of the recirculated air. This mixture then passes through an air washer having an apparatus dew point of 50.5°F. A part of the recirculated air (by-passed air), mixes with the air leaving the washer and the resulting mixture enters the fan which supplies air to the room at a temperature of 61°F. Neglect energy of the fan.

- Determine the tons of refrigeration required.
- Determine the volume of by-passed air in cfm.



$$67,500 - 14,400 = 53,100 \frac{\text{Btu}}{\text{HR}} \text{ sensible}$$

$$\text{SHR} = \frac{53,100}{67,500} = 0.789$$

LOAD RESULTS ~~ARE~~ WHEN

Total heat load from outside air (O/A IS LOWEST TO ROOM (NEUTRAL) CONDITIONS)

$$\begin{aligned} Q_{\text{TOTAL}} &= 4.5 \times \text{CFM} \times (h_B - h_A) \\ &= 4.5 \times 1200 \times (38.3 - 28.1) \\ &= 4.5 \times 1200 \times 10.2 \\ &= 55,100 \frac{\text{Btu}}{\text{HR.}} \end{aligned}$$

$$\begin{aligned} \text{Total heat load (room and outside air)} &= \begin{array}{r} 55,100 \\ 67,500 \\ \hline 122,600 \end{array} \frac{\text{Btu}}{\text{HR.}} \end{aligned}$$

$$\frac{122,600}{12,000} = 10.2 \text{ tons}$$

Supply Air Required

$$\begin{aligned} \text{CFM} &= \frac{Q_s}{1.08(t_A - t_E)} = \frac{53,100}{1.08(75 - 61)} \\ &= 3,520 \text{ CFM} \end{aligned}$$

Condition "D" air (leaving washer) must remove all heat gain in the store because the by-pass air cannot do any cooling. (same temperature)

$$\begin{aligned} \text{CFM (D, air)} &= \frac{53,100}{1.08(t_A - t_D)} \\ &= \frac{53,100}{1.08(75 - 50.2)} \\ &= 1,980 \text{ CFM} \end{aligned}$$

By-passed air = supply air-washer air

$$= 3,520 - 1,980$$

$$= 1,540 \text{ CFM}$$

Another method of determining total heat load (room and outside air) using 1980 CFM for the flow thru the washer, find the condition of the air entering the washer. $1980 \text{ CFM} - 1200 \text{ CFM}$

$= 780 \text{ CFM of recirculated air.}$

$$\frac{780}{1980} \times 75 = 29.5$$

$$\frac{1200}{1980} \times 91 = \frac{55.1}{84.6} \text{ mixture dry bulb entering washer. From}$$

psychrometric chart mixture wet bulb = 70.4. Total heat load

removed by washer:

$$Q_{\text{Total}} = 4.5 \times \text{CFM} \times (h_c - h_d)$$

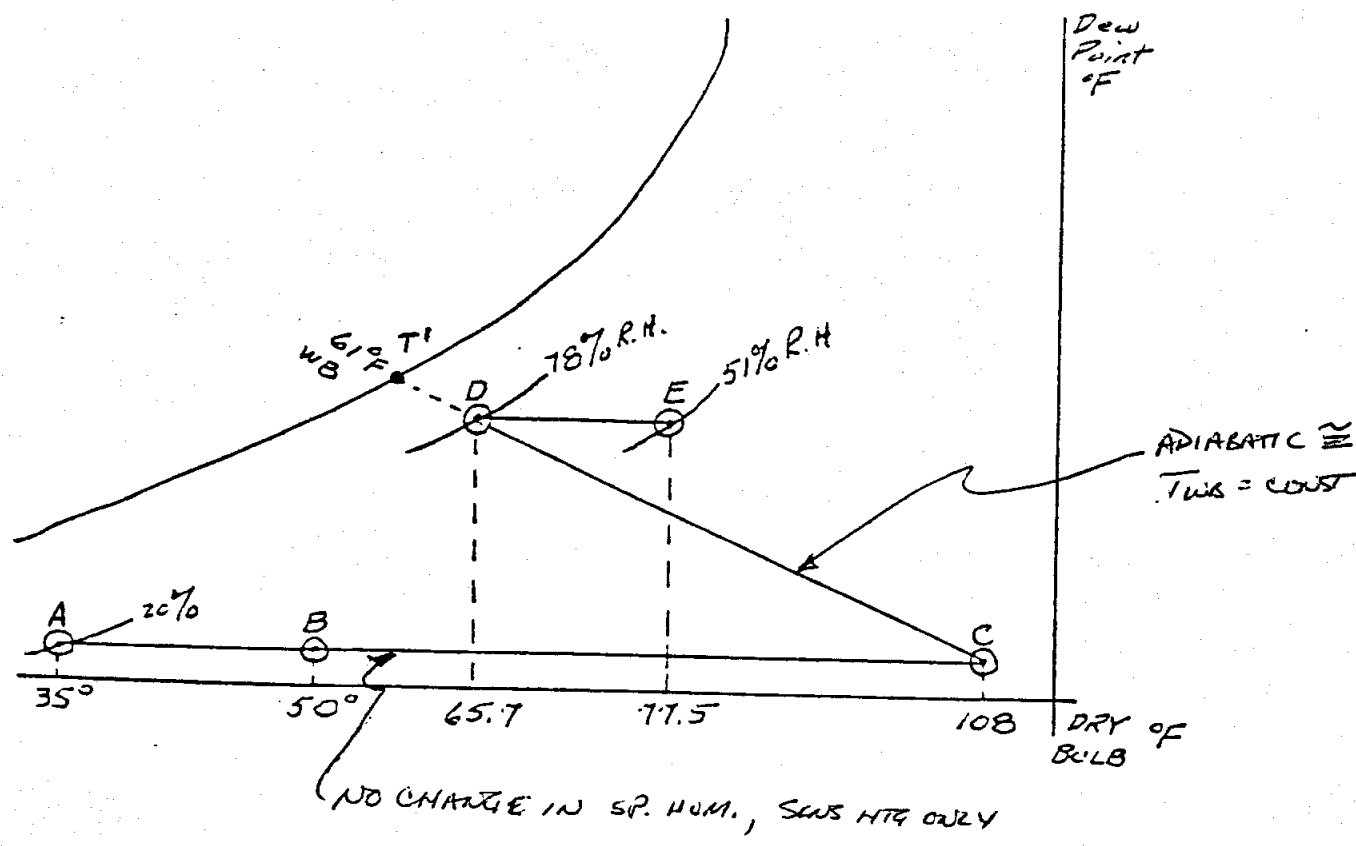
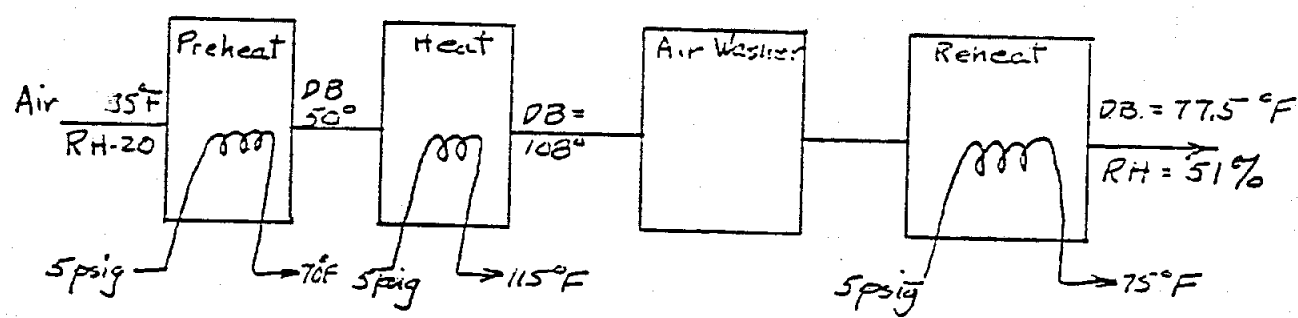
$$= 4.5 \times 1980 \times (34.2 - 20.5)$$

$$= 4.5 \times 1980 \times 13.7$$

$$= 122,200 \frac{\text{Btu}}{\text{hr.}}$$

10,000 cfm of air measured at 35°F dry-bulb temperature and a relative humidity of 20% enters a preheat coil and leaves at a dry bulb temperature of 50°F. The air then enters a heating coil and leaves at a dry-bulb temperature of 108°F. The air leaves the heating coil and enters an air washer using recirculated water whose saturation efficiency is 90%. The air leaves the air washer and enters a reheat coil. The air leaving the reheat coil is at 77.5°F dry-bulb (and 51% relative humidity). Steam to the heating coils is available 5 psig and saturated. The barometric pressure is 29.92 in Hg. The temperature of the condensate leaving the preheat coil is 70°F; leaving the heating coil is 115°F; and leaving the reheat coil is 75°F.

- Determine
- the lbm/hr of steam supplied to the preheat coil
 - The lbm/hr of steam supplied to the heating coil
 - the lbm/hr of steam supplied to the reheat coil
 - the gpm of make-up water to the air washer
 - the dry-bulb temperature and relative humidity of air leaving the air washer



$$\text{Saturation Efficiency} = \frac{t_c - t_D}{t_c - t^1}$$

Where

t_c = dry bulb temperature of entering air °F

t_D = dry bulb temperature of leaving air °F

t^1 = wet bulb temperature of entering air °F (SATURATION TEMP)

$$0.90 = \frac{108 - t_D}{108 - 61}$$

$$108 - t_D = 0.90 \times 47$$

$$- t_D = 42.3 - 108$$

$$t_D = 65.7 \text{ °F}$$

From Psychrometric Chart

$$h_A = 9.3 \frac{\text{Btu}}{\text{lb}_{\text{D.A.}}}$$

$$h_B = 12.9 \text{ "}$$

$$h_C = 26.8 \text{ "}$$

$$h_D = 27.1 \text{ "}$$

$$h_E = 30 \text{ "}$$

THERE IS REDUNDANT INFO GIVEN. STATE D IS DETERMINED KNOWING STATE C + SAT. EFF. OF WASHER. POINT E MUST BE ALONG A CONST SP. HUM. LINE; IF TDB_E IS SPECIFIED, THEN RH WILL RESULT

Heat added to air by preheat coil

$$\begin{aligned} Q &= 4.5 \text{ CFM } (h_B - h_A) \\ &= 4.5 \times 10,000 \times (12.9 - 9.3) \\ &= 45,000 \times 3.6 \\ &= 162,000 \frac{\text{Btu}}{\text{hr}} \end{aligned}$$

Note: We are assuming that the 10,000 CFM is at standard air conditions. If it was meant that the CFM should be strictly at condition A then CFM to be used would be

$$10,000 \times \frac{13.33}{12.48} = 10,700 \text{ CFM } \quad (\text{OR USE MASS FLOW})$$

$$\dot{m} = \frac{\text{CFM}}{v}$$

(a) lb_m/hr of steam supplied the preheat coil

$$\begin{aligned} \text{Steam enthalpy entering (5 psig} &= 20 \text{ psia)} = 1156.30 \\ \text{Steam enthalpy leaving (70°F)} &= \frac{38.04}{10} \frac{\text{Btu}}{\text{lb}} \\ &= 1118.26 \frac{\text{Btu}}{\text{lb}} \end{aligned}$$

$$\begin{aligned} \dot{Q} &= \dot{m} \Delta h \\ \dot{m} &= \frac{\dot{Q}}{\Delta h} = \frac{162,000}{1118.26} = 144.9 \frac{\text{lb}_m}{\text{hr}} \end{aligned}$$

Heat added to air by heating coil

$$\begin{aligned} Q &= 4.5 \text{ CFM } (h_C - h_B) \\ &= 4.5 \times 10,000 \times (26.8 - 12.9) \\ &= 45,000 \times 13.9 \\ &= 625,500 \frac{\text{Btu}}{\text{hr}} \end{aligned}$$

$$\text{NOTE: } h_{\text{SAT LIQ}} = T - 32^\circ$$

(b) lb_m/hr of steam supplied to the heating coil

$$\begin{aligned} \text{steam enthalpy entering} &= 1156.30 \\ \text{steam enthalpy leaving} &= 82.93 \\ \text{AT 115° SAT. LIQ.} &= 1073.37 \frac{\text{Btu}}{\text{lb}} \end{aligned} \quad \left. \vphantom{\begin{aligned} \text{steam enthalpy entering} \\ \text{steam enthalpy leaving} \end{aligned}} \right\} \text{FROM STEAM TABLES}$$

$$\begin{aligned} \dot{Q} &= \dot{m} \Delta h \\ \dot{m} &= \frac{\dot{Q}}{\Delta h} = \frac{625,500}{1073.37} = 582.7 \frac{\text{lb}_m}{\text{hr}} \end{aligned}$$

Heat added to air by reheat coil

$$\begin{aligned} Q &= 4.5 \text{ CFM } (h_E - h_D) \\ &= 4.5 \times 10,000 \times (30 - 27.1) \\ &= 45,000 \times 2.9 \\ &= 131,000 \text{ Btu/hr} \end{aligned}$$

(c) lb_m/hr of steam supplied the reheat coil

$$\begin{aligned} \text{steam enthalpy entering} &= 1156.30 \\ \text{steam enthalpy leaving} &= 43.03 \\ \text{AT 75° SAT. LIQ.} &= 1113.27 \frac{\text{Btu}}{\text{lb}} \end{aligned} \quad \left. \vphantom{\begin{aligned} \text{steam enthalpy entering} \\ \text{steam enthalpy leaving} \end{aligned}} \right\} \text{FROM STEAM TABLES}$$

$$\dot{m} = \frac{\dot{Q}}{\Delta h} = \frac{131,000}{1113.27} = 117.67 \frac{\text{lb.}}{\text{hr.}}$$

(d) The gpm of make up water to the air washer

$$\text{Humidity Ratio @ C} = 6 \text{ grains/lb. DA}$$

$$\text{Humidity Ratio @ D} = 73 \text{ grains/lb. DA}$$

$$\Delta \text{H.R.} = 67 \text{ grains/lb. DA}$$

$$\text{Moisture added to air} = \frac{4.5 \times \text{CFM} \times \Delta \text{H.R.}}{7000}$$

$$= \frac{4.5 \times 10,000 \times 67}{7000}$$

$$= 430 \text{ lb/hr}$$

$$\text{GPM} = \frac{430 \frac{\text{lb}}{\text{hr}}}{60 \frac{\text{min}}{\text{hr}} \times 8.33 \frac{\text{lb}}{\text{gal}}} = 0.859 \frac{\text{gal.}}{\text{min.}}$$

(e) From Psychrometric Chart

Dry Bulb Temp leaving washer 65.7 °F

Relative Humidity leaving washer 78%

$$4.5 = \frac{\text{FT}^3}{\text{MIN}} \times 60 \frac{\text{MIN}}{\text{HR}} \times .075 \frac{\text{LB}}{\text{FT}^3}$$

$$\text{Converts CFM} \rightarrow \frac{\text{LB}}{\text{HR}}$$

(b) Required air flow thru condenser

$$\begin{aligned} \text{CFM} &= \frac{Q \frac{\text{Btu}}{\text{hr}}}{4.5(h_B - h_A)} = \frac{5761 \times 60}{4.5(47.9 - 39.2)} \\ &= 8,850 \text{ CFM} \end{aligned}$$

(c) gpm of water evaporated by condenser

$$\text{H.R.}_A = 105 \frac{\text{grains}}{\text{lb}_{\text{DA}}}$$

$$\text{H.R.}_B = 175 \frac{\text{grains}}{\text{lb}_{\text{DA}}}$$

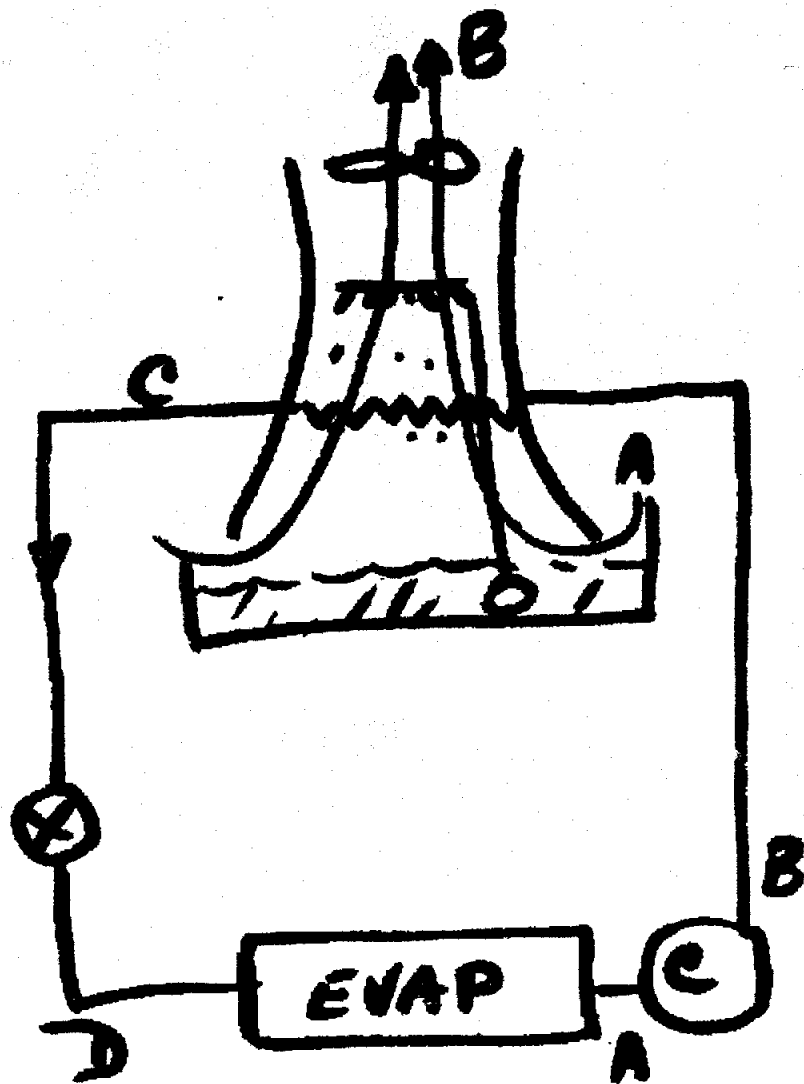
$$\text{Water Fbw} = \frac{4.5 \text{ CFM} (\text{H.R.}_B - \text{H.R.}_A)}{7000}$$

$$= \frac{4.5 \times 8850 \times 70}{7000}$$

$$= 398 \text{ lb/hr.}$$

$$= \frac{398 \frac{\text{lb}}{\text{hr}}}{8.33 \frac{\text{lb}}{\text{gal}}} \times \frac{1}{60 \frac{\text{min}}{\text{hr}}}$$

$$= 0.797 \frac{\text{gal.}}{\text{min.}}$$



1. *Phragmites* (Common Reed)

1ED

NCEE APRIL 1970

Sensible Heat Gain 50,000 Btu/h
Latent Heat Gain 30,000 Btu/h
Total Heat Gain 80,000 Btu/h

$$\text{Sensible Heat Ratio} = \frac{50,000}{80,000} = 0.625$$

$$\text{C.F.M. of Supply Air} = \frac{Q_s}{1.08 \Delta t} = \frac{50,000}{1.08(80-62)} = \frac{50,000}{1.08 \times 18} = 2,570 \text{ CFM}$$

Finding mixed air conditions (c)

$$\frac{1000}{2570} \times 95 = 37.0$$

$$\frac{1570}{2570} \times 80 = 48.9$$

85.9 °F D.B. of air entering coil (c)

Cooling Coil Enthalpy Change

$$\begin{aligned} \text{leaving cooling coil} &= 35 \frac{\text{Btu}}{\text{lb.}} \\ \text{entering cooling coil} &= 22 \frac{\text{Btu}}{\text{lb.}} \\ \Delta h &= 13 \frac{\text{Btu}}{\text{lb.}} \end{aligned}$$

$$\begin{aligned} \text{Cooling Coil Capacity} &= 4.5 \times \text{CFM} \times \Delta h \\ &= 4.5 \times 2,570 \times 13 \\ &= 150,200 \frac{\text{Btu}}{\text{HR}} \text{ Heat Removed} \end{aligned}$$

Reheat Coil Enthalpy Change

$$\begin{aligned} \text{leaving reheat coil} &= 24.1 \frac{\text{Btu}}{\text{lb.}} \\ \text{entering reheat coil} &= 22.0 \frac{\text{Btu}}{\text{lb.}} \\ \Delta h &= 2.1 \frac{\text{Btu}}{\text{lb.}} \end{aligned}$$

$$\begin{aligned} \text{Reheat Coil Capacity} &= 4.5 \times \text{CFM} \times \Delta h \\ &= 4.5 \times 2570 \times 2.1 \\ &= 24,300 \frac{\text{Btu}}{\text{HR}} \text{ Heat Added} \end{aligned}$$

To Verify Results

Outside Air Load:

$$h_{\text{outside conditions}} = 41.4 \frac{\text{Btu}}{\text{lb.}}$$

$$h_{\text{room conditions}} = 31.2 \frac{\text{Btu}}{\text{lb.}}$$

$$\Delta h = 10.2 \frac{\text{Btu}}{\text{lb.}}$$

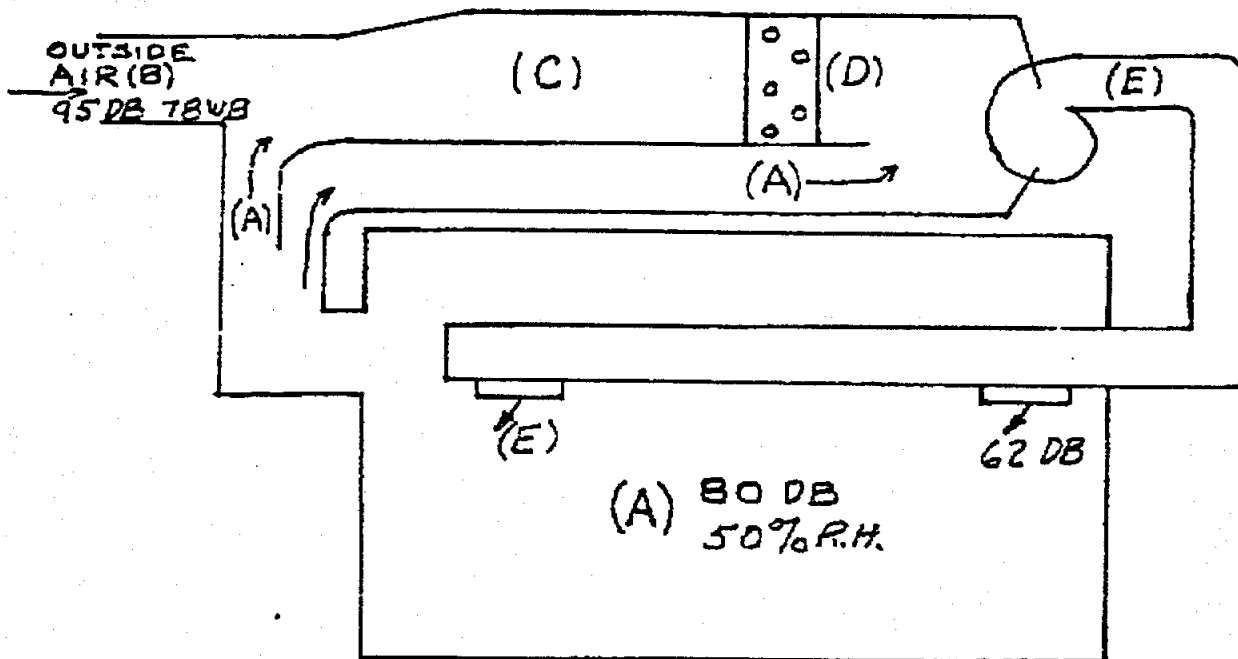
$$\begin{aligned} \text{Outside Air Load} &= 4.5 \times \text{CFM} \times \Delta h \\ &= 4.5 \times 1000 \times 10.2 \\ &= 45,900 \frac{\text{Btu}}{\text{HR}} \end{aligned}$$

Total Cooling Load

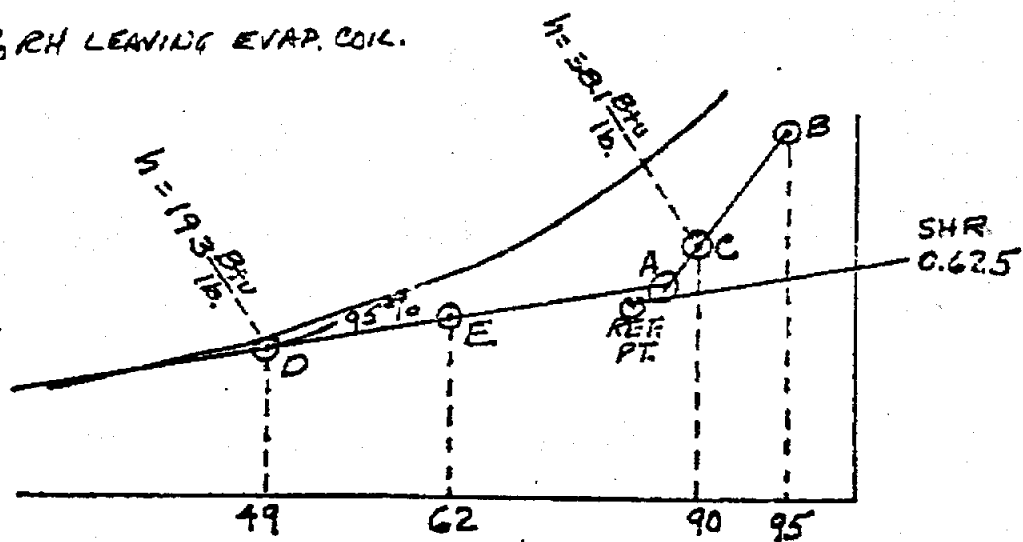
Outside Air	45,900	$\frac{\text{Btu}}{\text{HR}}$
Room	80,000	$\frac{\text{Btu}}{\text{HR}}$
Reheat	24,300	$\frac{\text{Btu}}{\text{HR}}$
	<u>150,200</u>	$\frac{\text{Btu}}{\text{HR}}$

NCEE APRIL 1970

SOLUTION WITHOUT REHEAT (USING BYPASS AIR)



ASSUMED 95% RH LEAVING EVAP. COIL.



NOTE:

ALL O/A GOES THRU COIL

IF SHR IS VERY LOW, THIS MAY NOT BE POSSIBLE

NCEE APRIL 1970

SOLUTION WITHOUT REHEAT

CFM THRU COIL

$$\begin{aligned} \text{CFM} &= \frac{Q_s}{1.08 \Delta t} = \frac{50,000}{1.08(80 - 49)} \\ &= 1493 \text{ CFM} \quad (T_A - T_D) \end{aligned}$$

FINDING MIXED AIR CONDITION

$$\frac{1000}{1493} \times 95 = 63.6 \quad (C)$$

$$\frac{493}{1493} \times 80 = \frac{26.4}{90.0} \text{ D.B.}$$

COOLING COIL ENTHALPY CHANGE

$$h_{\text{entering coil (C)}} = 38.1$$

$$h_{\text{leaving coil (D)}} = \underline{19.3}$$

$$\Delta h = 18.8 \text{ Btu/lb.}$$

$$\begin{aligned} \text{COILING COIL CAPACITY} &= 4.5 \times \text{CFM} \times \Delta h \\ &= 4.5 \times 1493 \times 18.8 \\ &= 126,307 \text{ Btu/hr.} \end{aligned}$$

To Verify Results

$$\text{Outside Air } Q_T = 45,900 \text{ (previous calculation)}$$

$$\text{Room } Q_T = \underline{80,000}$$

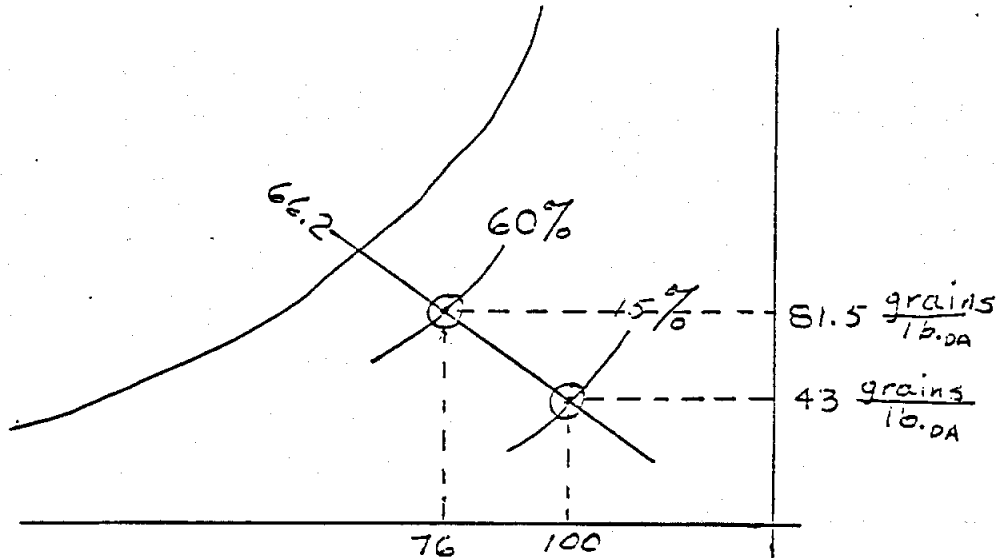
$$\text{Total } Q_T = 125,900$$

SAME AS BEFORE WITHOUT THE

R/H

NCEE DEC. 1967

The outside air temperature is 100 degrees F at 15% R.H. It is desired to cool 1500 CFM to the lowest possible temperature by evaporative cooling with the R.H. increasing to 60%. How many pounds of water per minute is required and what is the final temperature?



The solution is based on 1500 CFM of standard air

(a) Pounds of water per minute

$$\begin{aligned} \dot{m}_{H_2O} &= \frac{4.5 \times \text{CFM} (HR_2 - HR_1)}{7000} \\ &= \frac{4.5 \times 1500 (81.5 - 43)}{7000} \\ &= 37.125 \frac{\text{lb}}{\text{hr.}} \text{ or } 0.618 \frac{\text{lb.}}{\text{min.}} \end{aligned}$$

(b) Final temp

76°F (from chart)

LOAD CALCULATION PROBLEM

3/01 Kkh

A 10,000 FT² CHURCH SEATS 1000 PEOPLE AND IS TO COOLED TO INDOOR DESIGN CONDITIONS OF 76°F DB AND 64° WB. THE TRANSMISSION LOAD INCLUDING SOLAR LOADING IS 400,000 BTU/HR. LIGHTING AVERAGES 3 WATTS/FT² AND IS A MIXTURE OF INCANDESCENT AND FLUORESCENT.

OUTDOOR AIR DESIGN CONDITIONS ARE 95°F DB AND 76°F WB. FRESH AIR IS TO BE PROVIDED AT NORMAL ^{*}THEATER VENTILATION RATES.

CALCULATE :

- A. SENSIBLE, LATENT AND TOTAL SPACE COOLING LOADS
- B. TOTAL LOAD ON THE COOLING COIL AND COIL SHR

SOLUTION

A. SPACE COOLING LOAD

1. SENSIBLE

TRANSMISSION (GIVEN)

400,000 BTU/HR

LIGHTING - ASSUME $\frac{1}{2}$ INCANDESCENT

AND THAT BALLAST IS IN CONDITIONED SPACE

$$\begin{aligned}\text{LIGHTS} &= 3 \times \frac{10000}{2} \times 3.413 + 3 \times \frac{10000}{2} \times 3.413 \times 1.2 \\ &= 112600\end{aligned}$$

PEOPLE - ASSUME AVERAGE GROUPING AT REST

$$Q_s = 1000 \times 230 = 230000$$

$$\text{SENSIBLE} = 400000 + 112600 + 230000 = 1372600 \text{ BTU/HR}$$

2. LATENT

$$\text{PEOPLE } Q_L = 1000 \times 120 = 120000$$

$$3. \text{ TOTAL} = 1372600 + 120000 = 1492600 \text{ BTU/HR}$$

$$\text{SHR} = \frac{1372600}{1492600} = .92 = 124.4 \text{ TONS}$$

B. TOTAL LOAD ON COIL

ADDITIONAL LOAD IS DUE TO REQUIRED VENTILATION AIR

$$\text{CFM} = 1000 \times 15 \text{ CFM/PERSON} = 15000$$

$$\underline{Q_{\text{SENS}}} = \text{CFM} \times 1.08 (T_{\text{OA}} - T_{\text{DES}})$$

$$= 15000 \times 1.08 (95 - 76)$$

$$= 307800 \text{ BTU/HR}$$

TO FIND OUTSIDE AIR LATENT LOAD, THE SPECIFIC HUMIDITIES MUST BE FOUND FROM A PSYCH. CHART

$$@ 76/64 \quad W = 70 \text{ gr/lb}$$

$$@ 95/76 \quad W = 105 \text{ gr/lb}$$

$$Q_{LAT} = CFM \times .68 \times (W_{OA} - W_{RA})$$

$$= 15000 \times .68 (105 - 70) = 357000 \text{ BTU/HR}$$

TOTAL COIL LOAD

$$Q_{SENS} = 1372600 + 307800 = 1680400$$

$$Q_{LAT} = 120000 + 357000 = \frac{477000}{2157400} \text{ BTU/HR}$$

$$Q_{TOT} =$$

$$2157400 \text{ BTU/HR}$$

$$179.8 \text{ TONS}$$

$$SHR = \frac{1680400}{2157400} = .78$$

PE Review- Air Conditioning

A practical method to establish a Sensible Heat Ratio (SHR) line on a psychometric chart.

Background: The SHR line indicates the loci of points along which conditioned air must be delivered to a conditioned space (in proper quantity) to match the design load and hence produce the desired room conditions.

$$SHR = Q_{sens} / Q_{total} = Q_{sens} / (Q_{sens} + Q_{latent})$$

Using simplified approximate equations:

$$Q_{sens} = 1.08 * CFM * \Delta T_{db}$$

$$\begin{aligned} Q_{latent} &= .68 * CFM * \Delta w \text{ (grains / lb)} \\ &= 4760 * CFM * \Delta w \text{ (lb / lb)} \end{aligned}$$

$$SHR = \frac{1.08 * CFM * \Delta T_{db}}{1.08 * CFM * \Delta T_{db} + .68 * CFM * \Delta w}$$

Rearranging:

$$\Delta w = \frac{(1 - SHR) * 1.08 * \Delta T_{db}}{.68 * SHR} \quad (\text{gr/lb})$$

$$= \frac{(1 - SHR) * 1.08 * \Delta T_{db}}{4760 * SHR} \quad (\text{lb/lb})$$

Procedure:

1. Plot point A, the known design conditions.
2. For a convenient increment of ΔT_{db} (say 10 or 20 deg), calculate Δw with the known SHR.
3. Calculate conditions for point B:

$$T_{db} = T_{Adb} + \Delta T_{db}$$

$$w = w_A + \Delta w$$

4. Plot point B
5. Connect points A and B to establish the SHR line.

Example:

Design conditions: $T_{db}=78$ degF, RH = 50%, SHR=.8

From psych chart : $w = 71.5$ gr/lb (.01022 lb/lb)

For:

$$\Delta T = 20 \text{ deg}$$

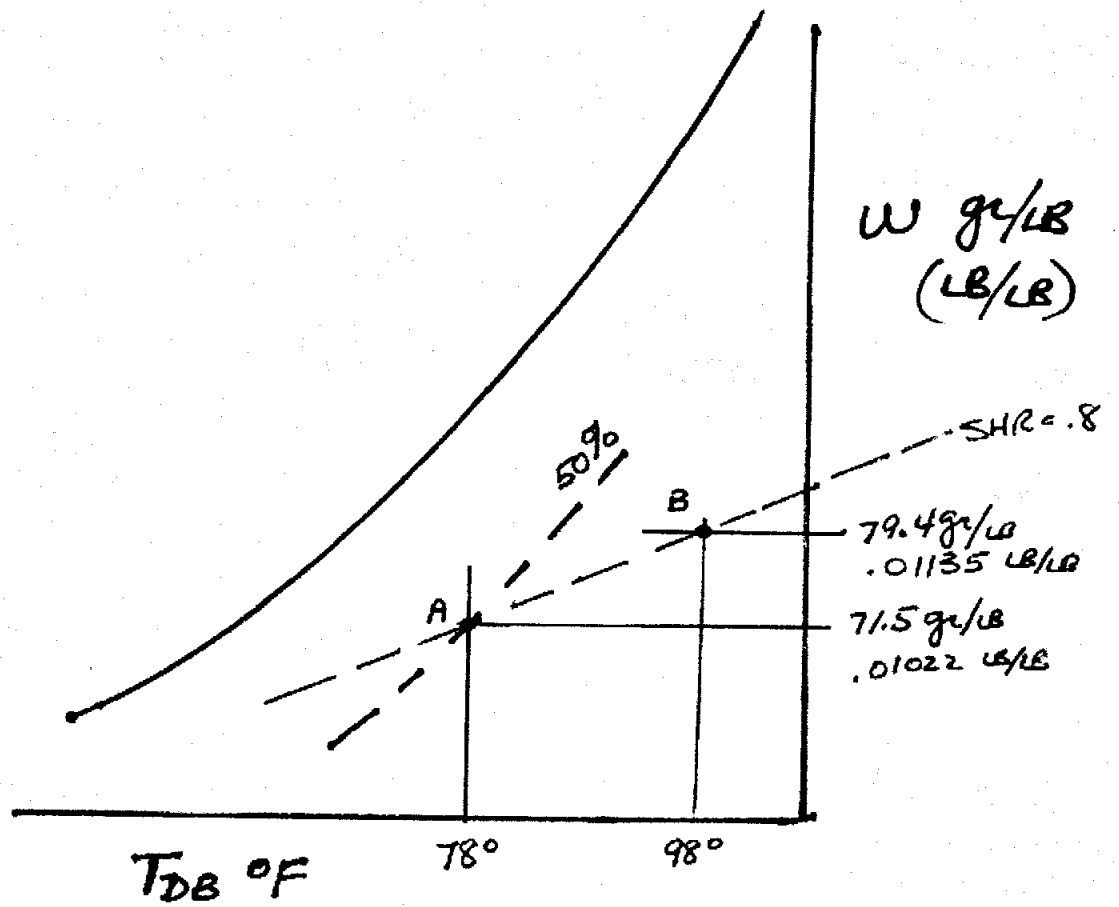
$$\Delta w = \frac{(1-.8) * 1.08 * 20}{.68 * .8} = 7.9 \text{ gr / lb } (.001134 \text{ lb/lb})$$

Therefor:

$$T_{db} = 78 + 20 = 98 \text{ degF}$$

$$w = 71.5 + 7.9 = 79.4 \text{ gr / lb } (.01135 \text{ lb/lb})$$

Since the selected design point is the "bulls eye" on the Trane Psych chart in this case we can check to see where the SHR line falls on the SHR scale.

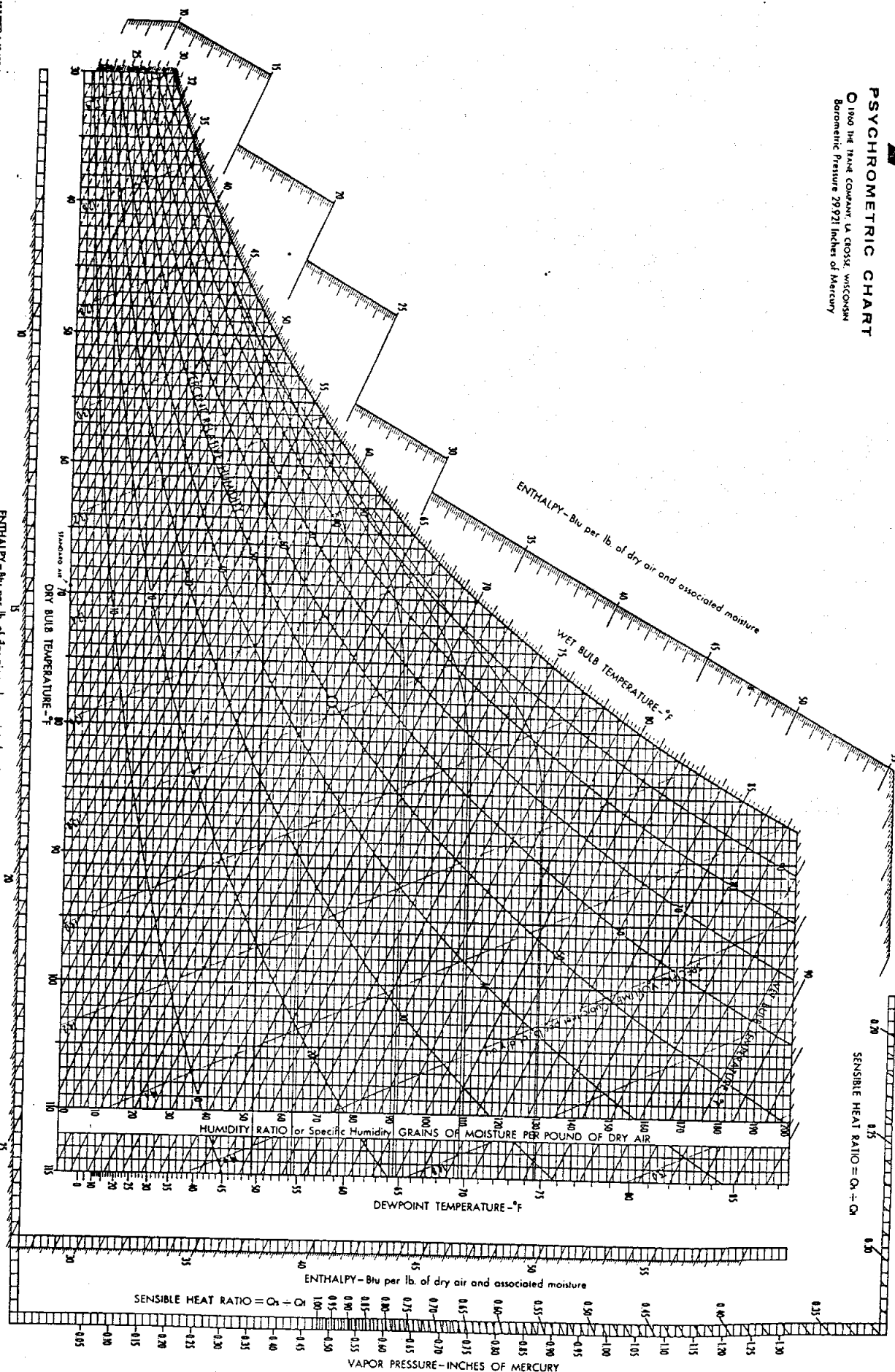




PSYCHROMETRIC CHART
© 1960 THE TRANE COMPANY, LA CROSSE, WISCONSIN
Barometric Pressure 29.921 inches of Mercury

FORM 14-6000 1-45 1959 Rev. 1963
Supersedes 14-600 (11/17/57 Edition)

ENTHALPY - Btu per lb. of dry air and associated moisture



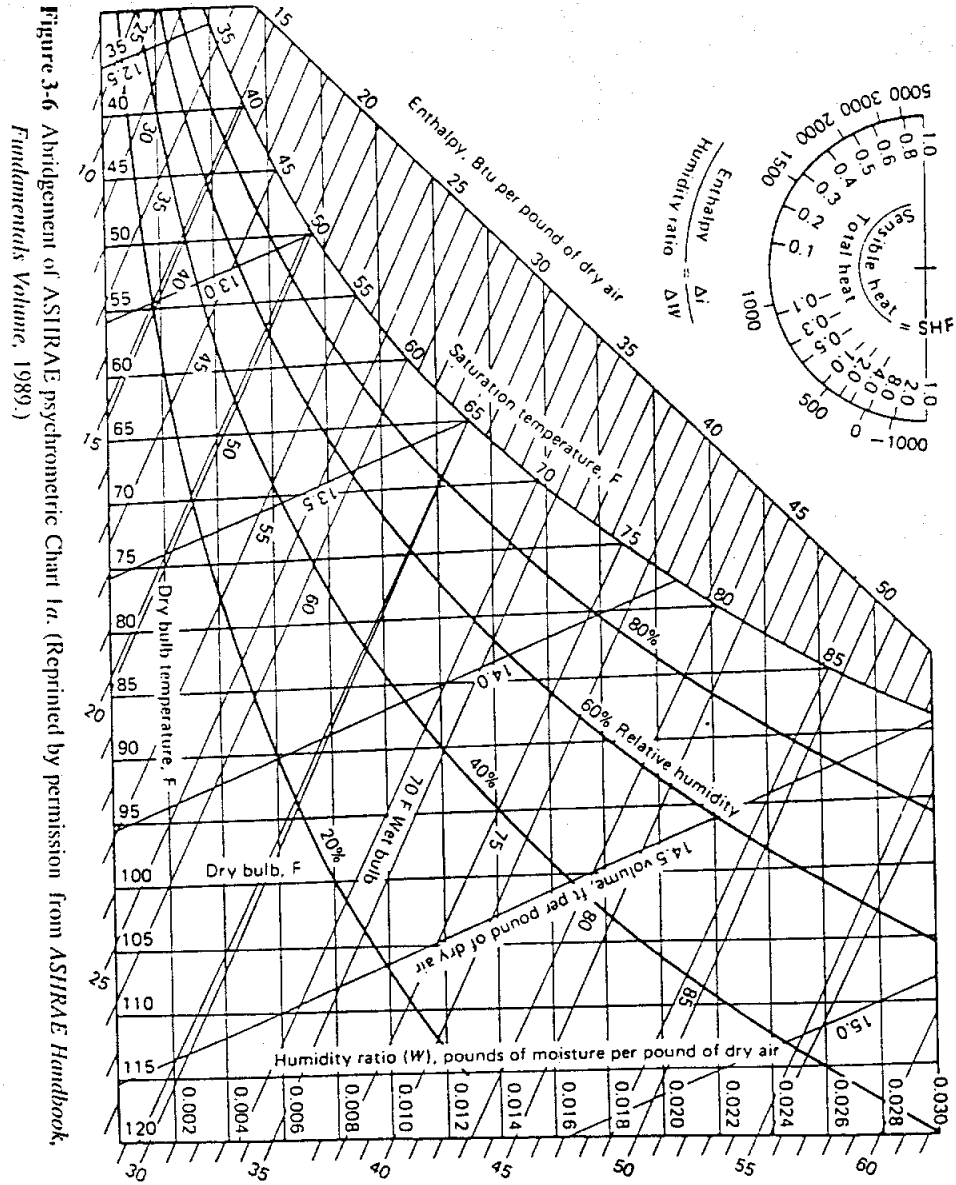


Figure 3-6 Abridgement of ASHRAE psychrometric Chart 1a. (Reprinted by permission from ASHRAE Handbook, Fundamentals Volume, 1989.)

ASHRAE PSYCHROMETRIC CHART NO. 1

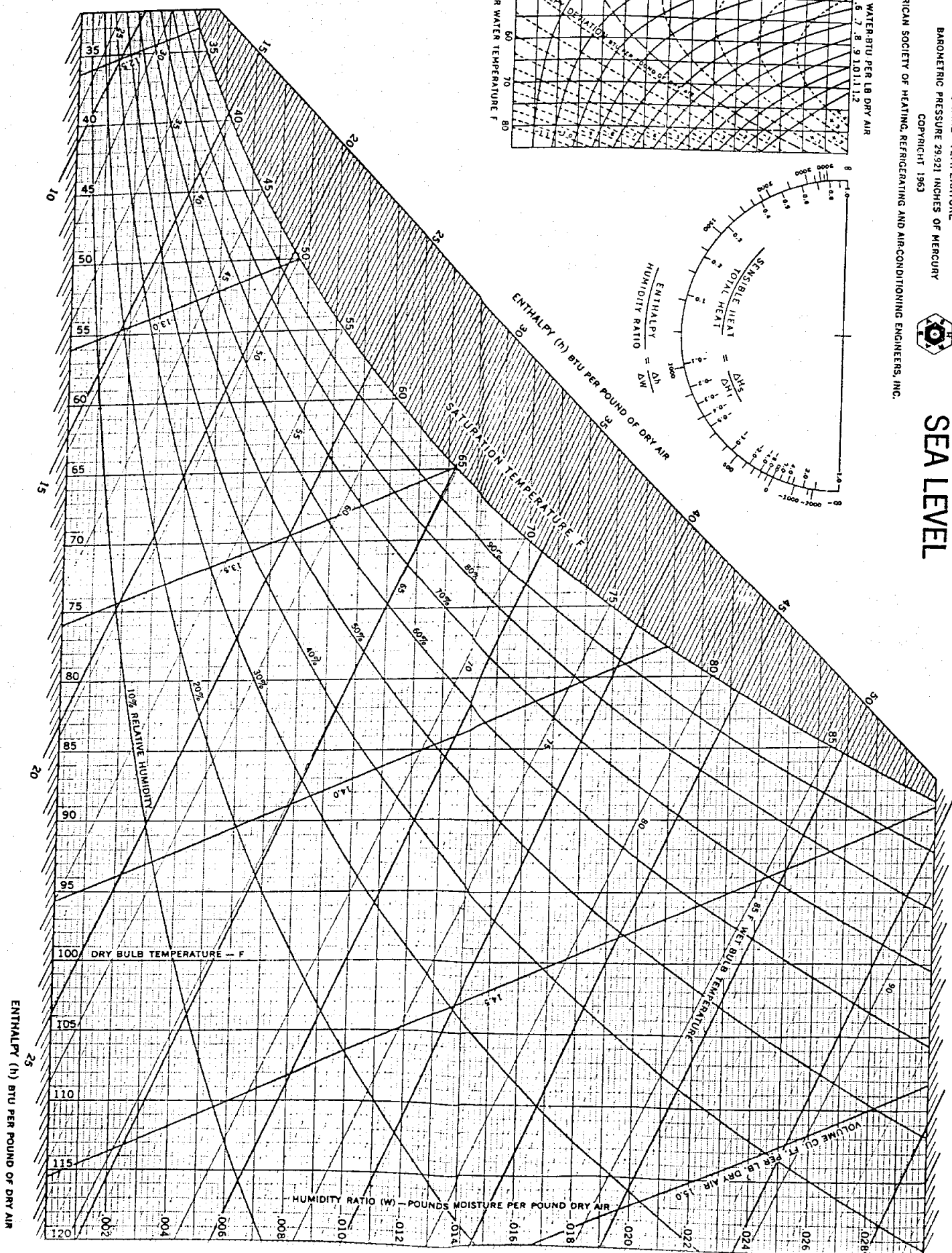
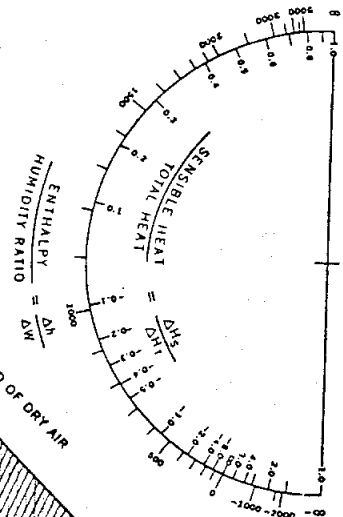
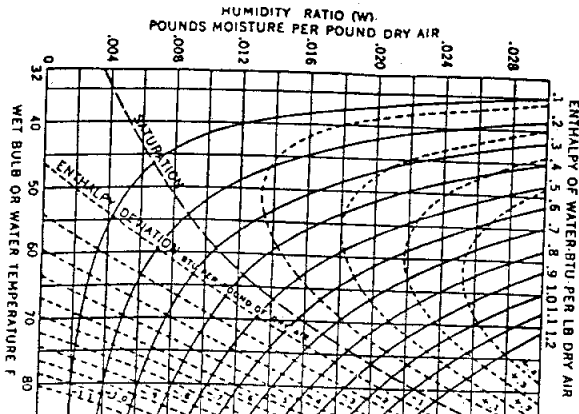
Chart 1a

NORMAL TEMPERATURE
BAROMETRIC PRESSURE 29.921 INCHES OF MERCURY
COPYRIGHT 1963

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.



SEA LEVEL



ASHRAE PSYCHROMETRIC CHART NO. 1

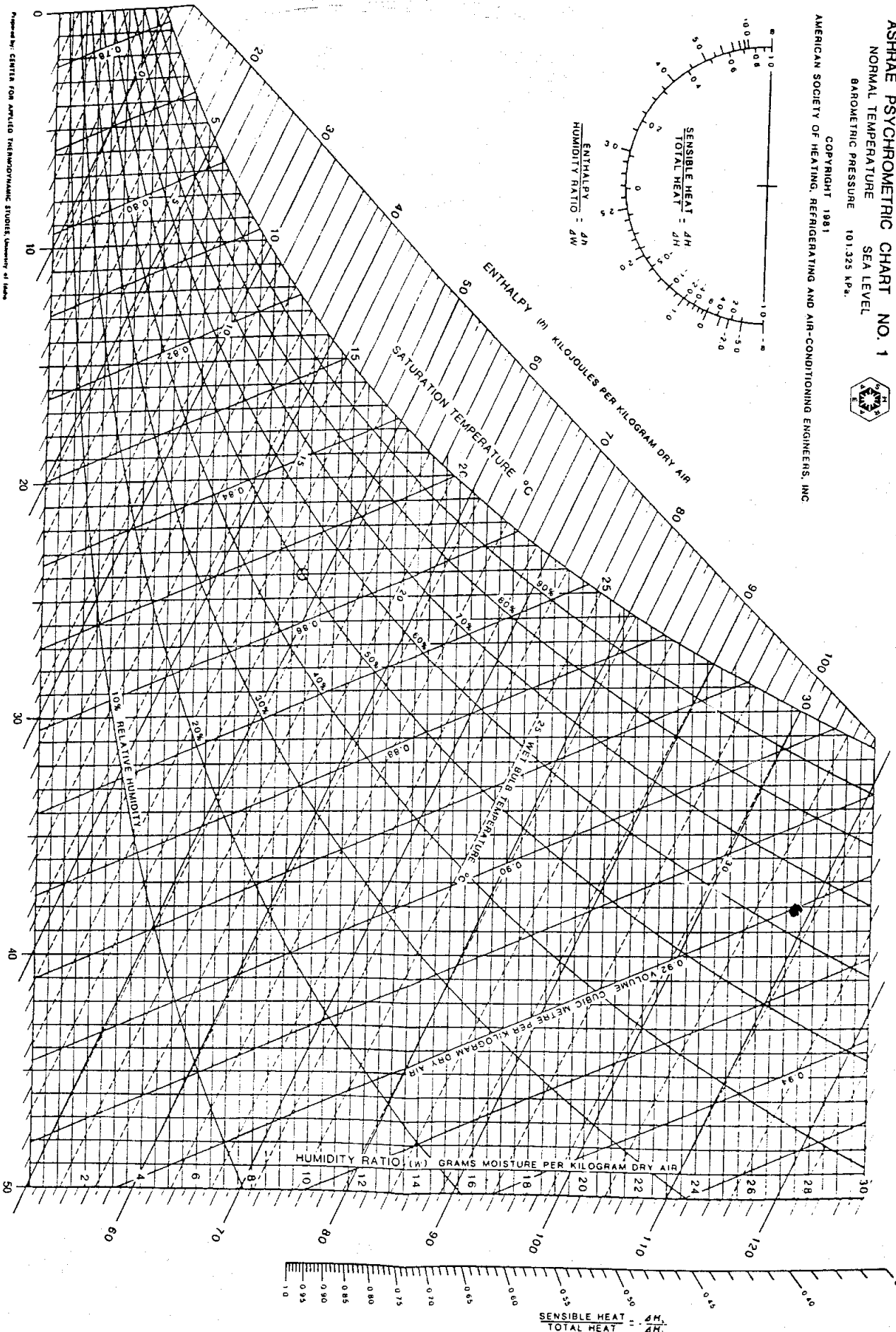
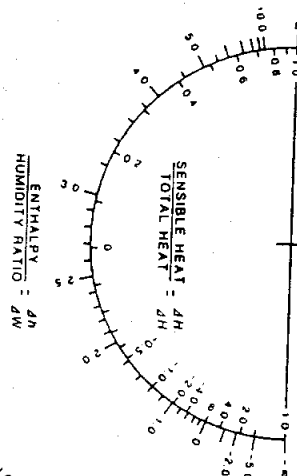
NORMAL TEMPERATURE

SEA LEVEL

BAROMETRIC PRESSURE 101.325 kPa

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P.E. REFRESHER - REFRIGERATION

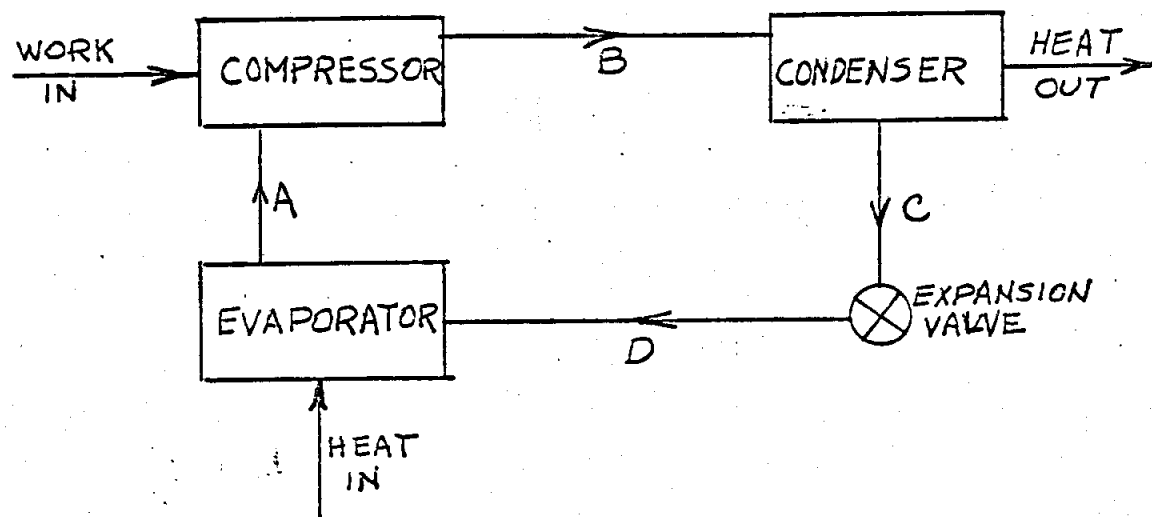


Fig. 1 Refrigeration System

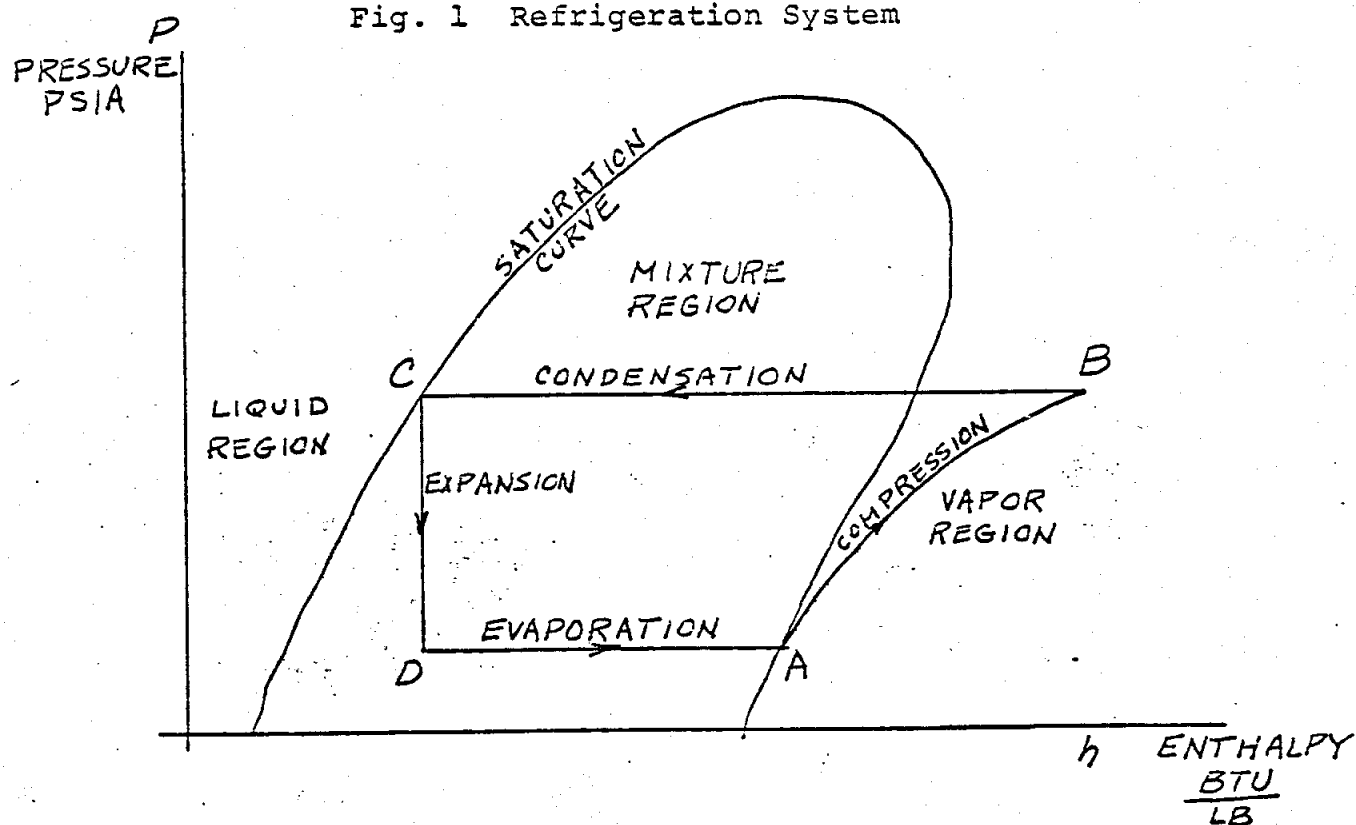


Fig. 2 Pressure - Enthalpy Chart

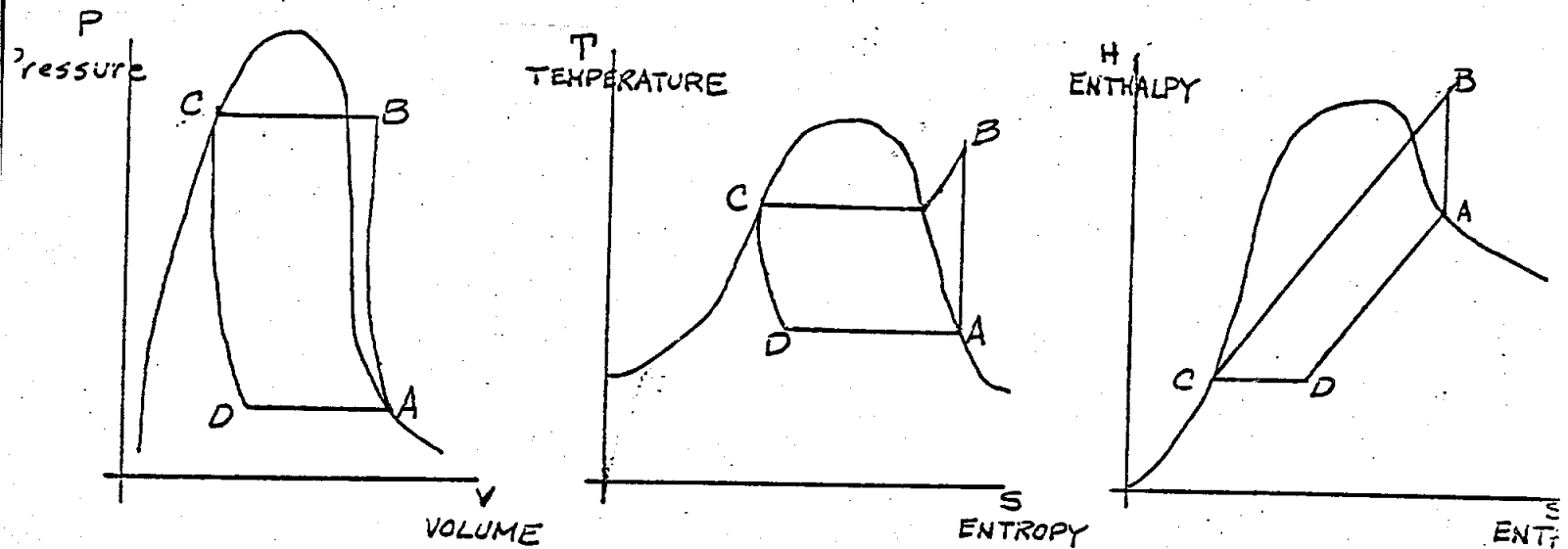


Fig. 3 Refrigeration Cycle on Other Commonly-Used Charts

First Law of Thermodynamics (expressed in Btu/lbm)

$$Q + h_1 + \frac{V_1^2}{2g_0J} + \frac{Z_1g}{g_0J} = W_x + h_2 + \frac{V_2^2}{2g_0J} + \frac{Z_2g}{g_0J}$$

Omitting Kinetic and Potential Energy Terms

For Compressor (neglecting heat loss)

$$h_A = W_x + h_B \quad \text{or} \quad W_x = h_A - h_B$$

$$\text{or } \dot{W}_x \frac{\text{Btu}}{\text{hr}} = \dot{m} \frac{\text{lb}}{\text{hr}} (h_A - h_B) \frac{\text{Btu}}{\text{lb}}$$

For Condenser

$$Q + h_B = h_C \quad \text{or} \quad Q = h_C - h_B$$

$$\text{or } \dot{Q} \frac{\text{Btu}}{\text{hr}} = \dot{m} \frac{\text{lb}}{\text{hr}} (h_C - h_B) \frac{\text{Btu}}{\text{lb}}$$

For Evaporator

$$Q + h_D = h_A \quad \text{or} \quad Q = h_A - h_D$$

$$\text{or } \dot{Q} \frac{\text{Btu}}{\text{hr}} = \dot{m} \frac{\text{lb}}{\text{hr}} (h_A - h_D) \frac{\text{Btu}}{\text{lb}}$$

If the refrigerating capacity of the system is known and the conditions of the refrigerant entering and leaving the evaporator are known the mass flow rate of the refrigerant is found by

$$\dot{m} = \frac{\dot{Q}_{evap}}{h_A - h_D}$$

This flow rate is the same for all components of the system. The condenser heat rejection can be found by

$$\dot{Q}_{COND} = \dot{m} (h_C - h_B)$$

The compressor horse power is found by

$$\dot{W} = \dot{m} (h_A - h_B) \text{ times appropriate conversion factor}$$

Definitions:

Degrees of superheat-The number of degrees above saturation temperature.

Degrees of Subcooling-The number of degrees below saturation temperature.

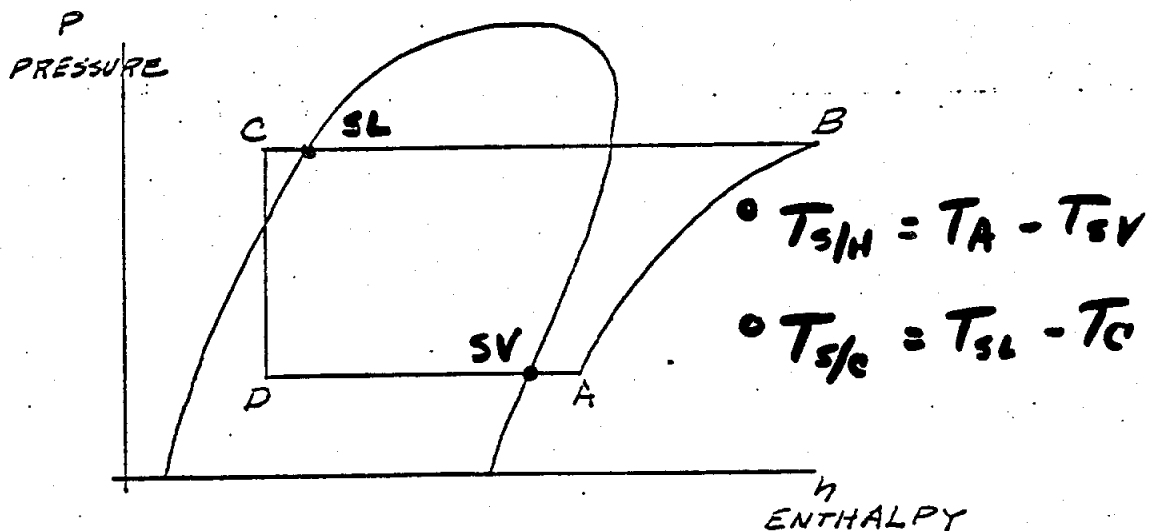


Fig. 4 Practical Refrigeration Cycle

Because constant temperature lines are almost vertical in the liquid region the enthalpy of subcooled refrigerant can be found by using saturated liquid enthalpy values. at the refrigerant temperature.

A Freon-12 vapor compression refrigeration system has a compressor of 80% adiabatic efficiency. At inlet to the compressor, the refrigerant is at a pressure of 5.3 psig and has 40 degrees superheat; at discharge from the compressor, the pressure is 100 psia.

Saturated liquid Freon at 100 psia leaves the condenser and passes through an expansion valve into the evaporator coils. The condenser is supplied with cooling water which enters at 60 degrees F and which leaves at 70 degrees F. For a plant capacity of five tons, find:

- the horsepower required to drive the compressor if it has a mechanical efficiency of 92%.
- The required capacity in gpm of the cooling water pump.

From Faires: Thermodynamics

Adiabatic Compression Efficiency =

$\frac{\text{Theoretical Power (isentropic compression)}}{\text{Actual Indicated Power}}$

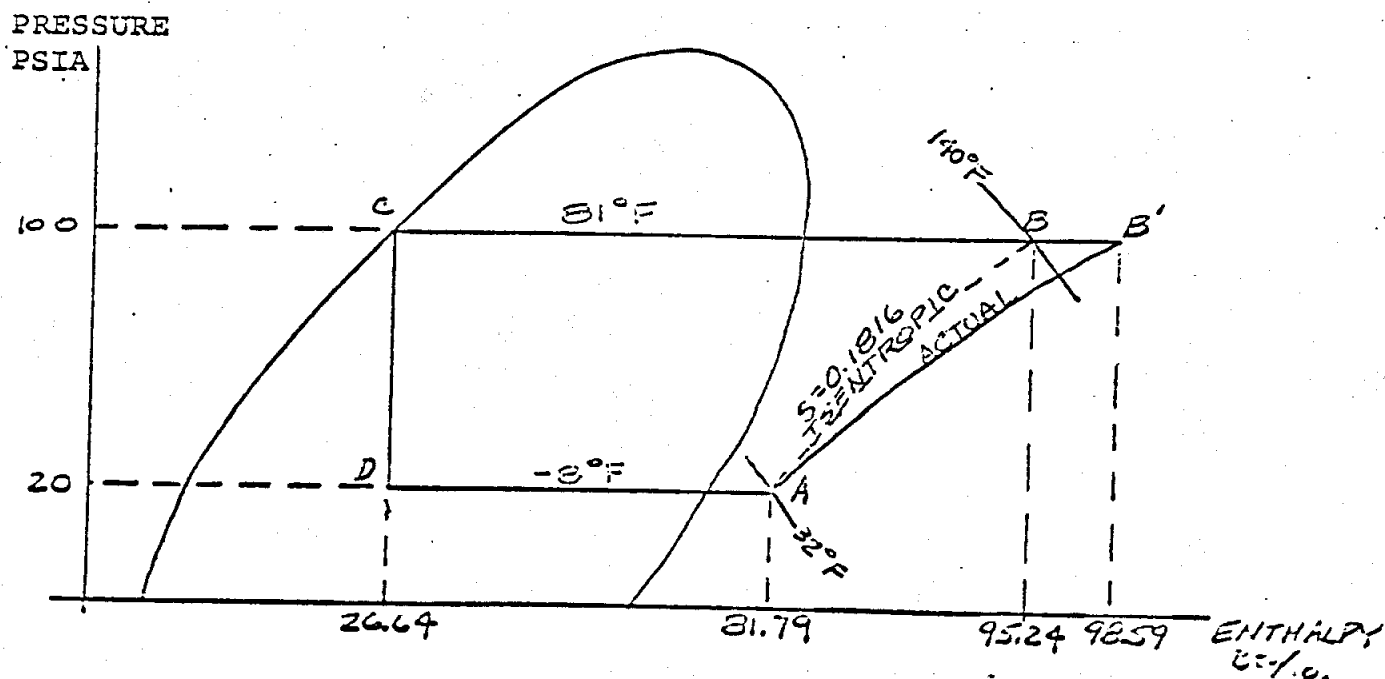
$$0.80 = \frac{h_B - h_A}{h_{B'} - h_A}$$

$$h_{B'} = \frac{h_B - h_A}{0.80} + h_A$$

$$= \frac{95.24 - 81.79}{0.80} + 81.79$$

$$= 98.59 \frac{\text{Btu}}{\text{lb.}}$$

$$5.3 \text{ psig} = (14.7 + 5.3) \text{ psia} = 20 \text{ psia}$$



a. Mass flow rate for 5 tons of refrigeration

$$\frac{5 \times 200}{h_A - h_D} = \frac{1000}{81.79 - 26.64} = 18.15 \frac{\text{lb}}{\text{min.}}$$

$$\frac{\text{lb}}{\text{min}} \times \frac{\text{Btu}}{\text{lb}} \times \left(\frac{778 \times}{60 \times 550} \right) = 0.02358$$

Theoretical Compressor Horsepower

$$\begin{aligned} \text{Theor. H.P.} &= \dot{m} (h_{B'} - h_A) \times 0.02356 \\ &= 18.15 (98.59 - 81.79) \times 0.02356 \\ &= 7.19 \text{ H.P.} \end{aligned}$$

$$\text{Actual Horsepower} = \frac{\text{THEORETICAL H.P.}}{\text{EFFICIENCY}} = \frac{7.19}{0.92} = 7.80 \text{ H.P.}$$

H Heat Rejection of Condenser

$$\begin{aligned} Q_{\text{COND}} &= \dot{m} (h_{B'} - h_C) \\ &= 18.15 (98.59 - 26.64) \\ &= 18.15 \times 71.95 \\ &= 1,308 \frac{\text{Btu}}{\text{min}} \end{aligned}$$

By the sensible heat relationship

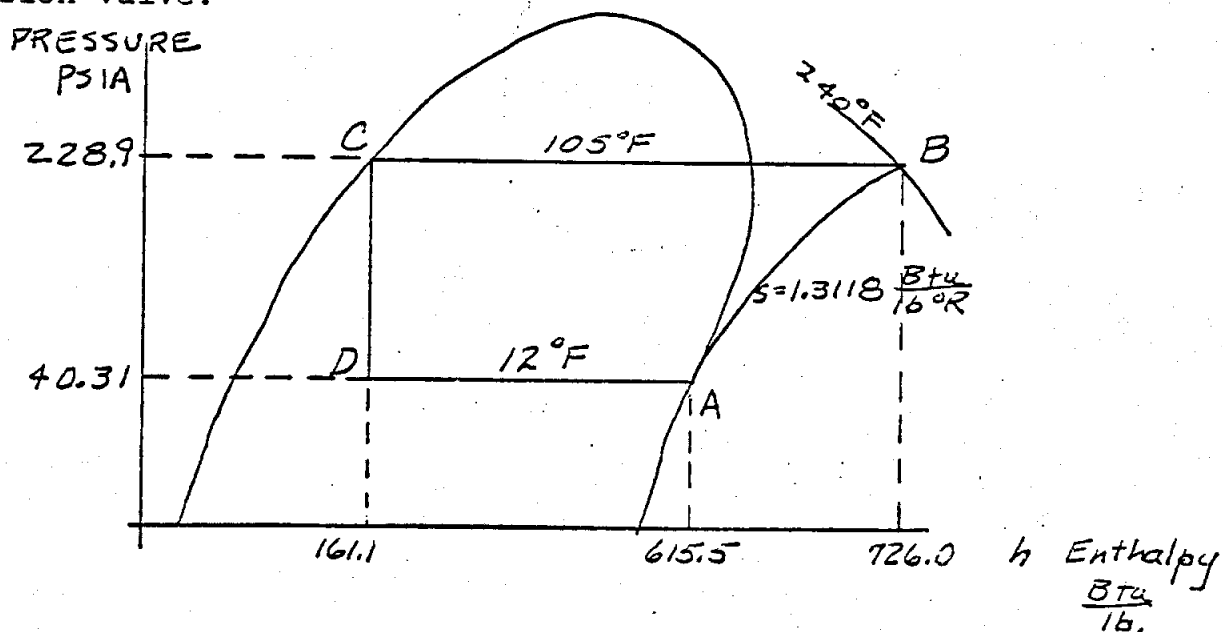
$$\dot{Q} = \dot{m} c \Delta t$$

$$\dot{m} = \frac{\dot{Q}}{c \Delta t} = \frac{1,308}{1.0 \times 10} = 130.8 \frac{\text{lb.}}{\text{min.}}$$

$$= \frac{130.8}{8.33} = 15.7 \frac{\text{gal.}}{\text{min.}}$$

$$0.02356 = \frac{778}{60 \times 550}$$

An ammonia refrigeration unit has an 180-rpm, double-acting compressor whose cylinder has a diameter of 8 in. with a 10-inch stroke and whose clearance is 5%. It is desired to calculate the capacity of the unit in tons of refrigeration at an evaporation temperature of 12°F and a condensing temperature of 105°F. Assume that the 12°F vapor is saturated as it enters the compressor. Also assume that the liquid ammonia is not subcooled at the entrance to the expansion valve.



$$\text{Theoretical Displacement} = 2 \times \frac{\pi \left(\frac{8}{12}\right)^2}{4} \times \frac{10}{12} \times 180 = 104.9 \frac{\text{ft.}^3}{\text{min}}$$

From: APPLICATION OF THERMODYNAMICS by BERNARD D. WOOD

Volumetric Efficiency

$$\eta_v = 1 + CF - CF \left(\frac{p_2}{p_1} \right)^{\frac{1}{n}} \quad \text{THEORETICAL VOLUMETRIC EFF.}$$

where $CF = \frac{\text{CLEARANCE VOLUME}}{\text{PISTON DISPLACEMENT}}$

From Mark's Handbook K for ammonia = 1.29 $K \approx n$ for this process

Clearance = $CF = 0.05$

$$\begin{aligned}\eta_r &= 1 + 0.05 - 0.05 \left(\frac{228.9}{40.31} \right)^{\frac{1}{1.29}} \\ &= 1.05 - 0.05 \times 5.68^{0.775} \\ &= 1.05 - 0.05 \times 3.84 \\ &= 1.05 - 0.192 \\ &= 0.858\end{aligned}$$

THEORETICAL

The actual volumetric flow rate is

$$104.9 \times 0.858 = 89.9 \frac{\text{ft}^3}{\text{min}}$$

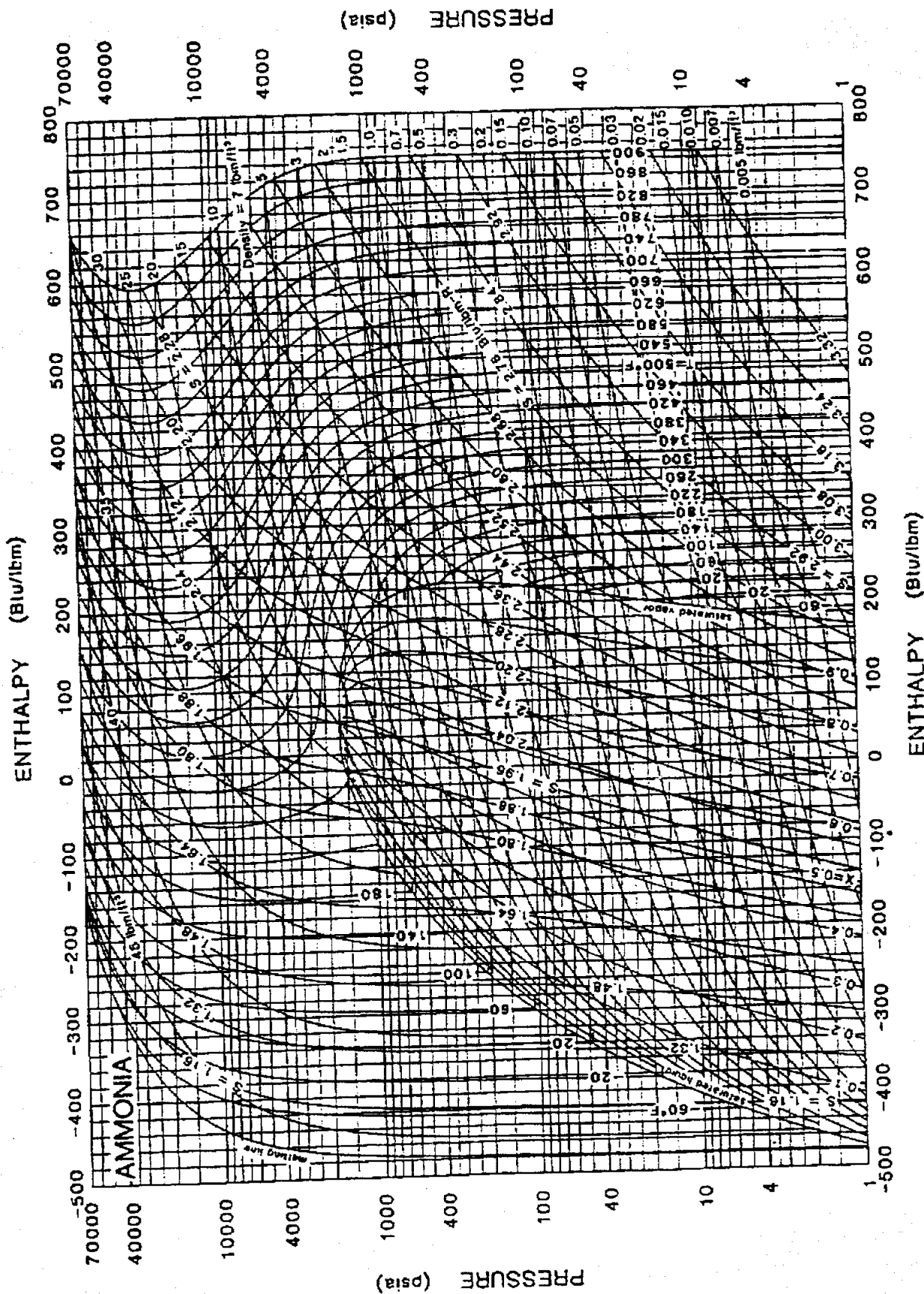
The mass flow rate is

$$\dot{m} = \frac{\dot{V}}{v} = \frac{89.9}{6.996} = 12.82 \frac{\text{lb.}}{\text{min.}}$$

The evaporator capacity is

$$\begin{aligned}\dot{Q} &= \dot{m}(h_A - h_D) \\ &= 12.82(615.5 - 161.1) \\ &= 12.82 \times 454.4 \\ &= 5,830 \frac{\text{BTU}}{\text{min}}\end{aligned}$$

$$\dot{Q} = \frac{5,830}{200} = 29.1 \text{ Tons}$$



Prepared by: CHEMISTRON SYSTEMS INCORPORATED, STURIS, UNIVERSITY OF ALABAMA
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(NCEE April 1971)

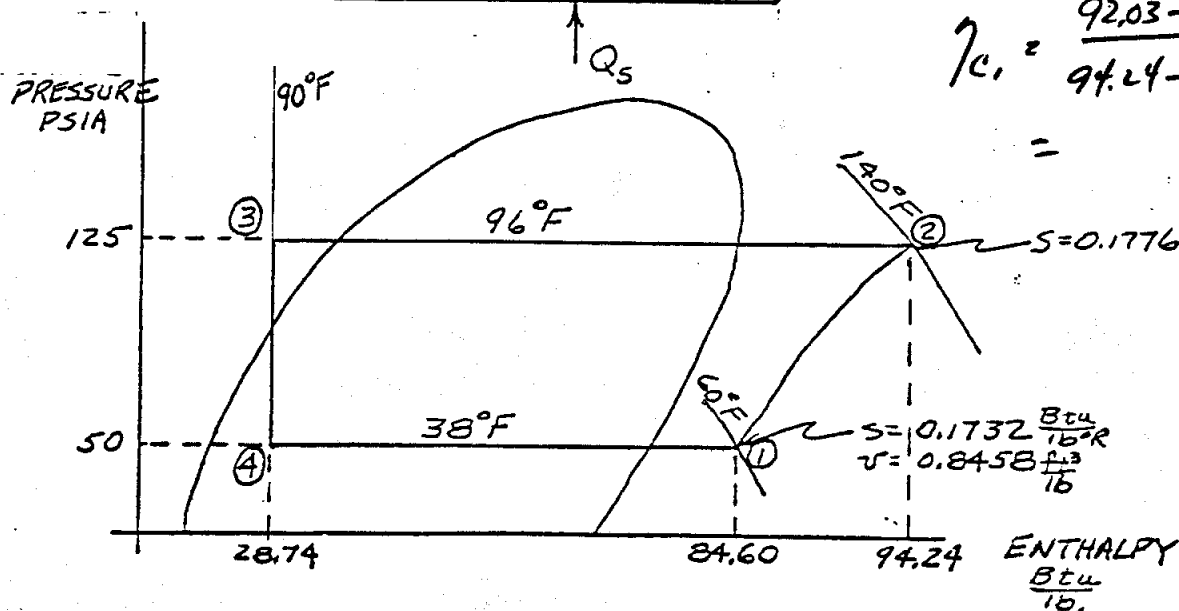
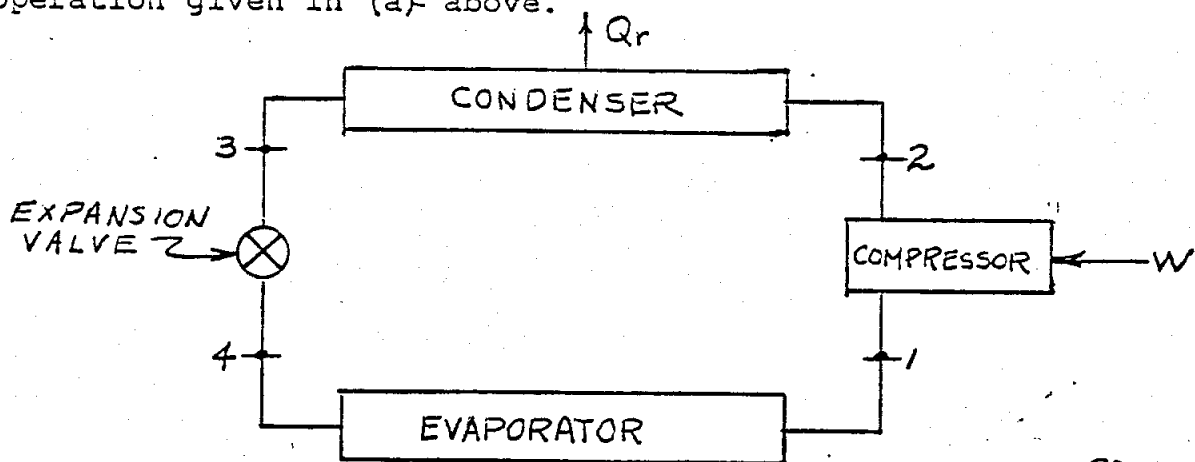
A given refrigeration system uses Freon-12 for the working medium and produces 50,000 Btu/hr cooling with the following operating conditions:

The compressor discharge pressure and temperature are 125 psia and 140°F, respectively.

The temperature of the Freon leaving the condenser is 90°F.

The pressure and temperature leaving the evaporator are 50 psia and 60°F, respectively.

- (a) If the system has a Positive Displacement constant speed compressor, and the cooling area conditions are changed to necessitate operation at a pressure of 35 psia and a temperature of 40°F in the evaporator, determine (by use of either chart or table values) the expected cooling capacity at the new condition of operation. Assume that the temperature of the Freon leaving the condenser and the discharge pressure leaving the compressor remain unchanged. (Please use station numbers as shown on sketch).
- (b) If the compression process is adiabatic and the rate of entropy increase remains essentially the same during compression, determine by calculations the C.O.P. for the original condition of operation and for the condition of operation given in (a) above.



mass Flow Rate:

$$\dot{m} = \frac{\dot{Q}}{h_1 - h_4} = \frac{50,000}{84.60 - 28.74}$$

$$= \frac{50,000}{55.86} = 894 \frac{\text{lb.}}{\text{hr.}}$$

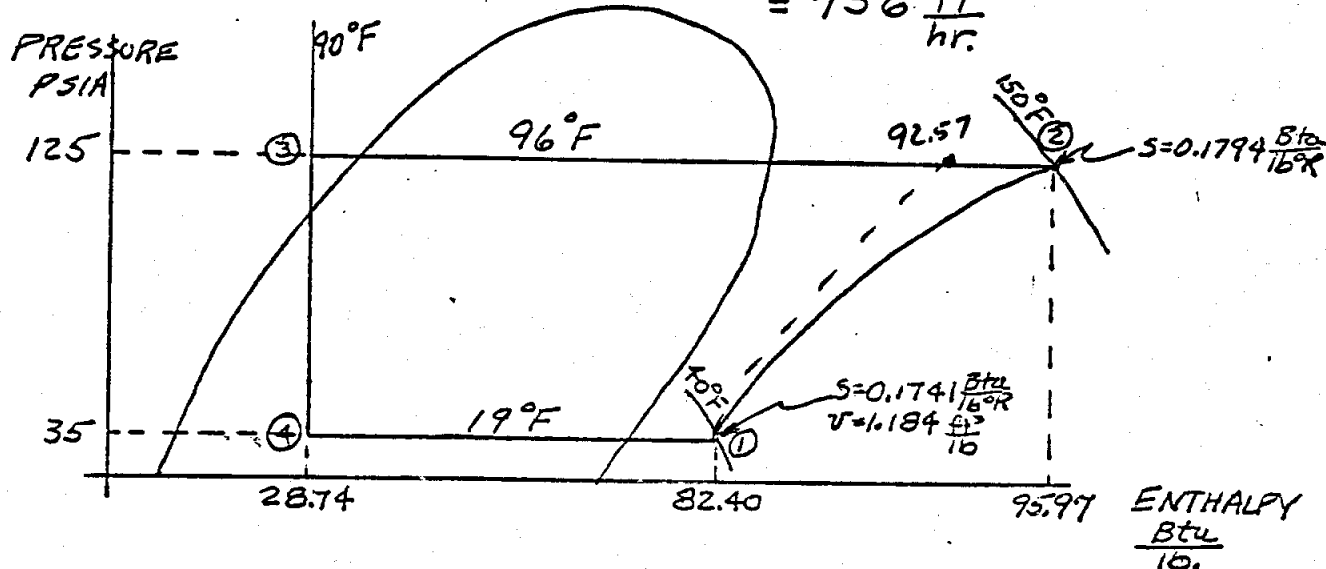
Volume Flow Rate:

$$\dot{V} \frac{\text{ft}^3}{\text{hr}} = \dot{m} \frac{\text{lb.}}{\text{hr.}} \cdot v_1 \frac{\text{ft}^3}{\text{lb.}}$$

$$= 894 \cdot 0.8458$$

$$= 756 \frac{\text{ft}^3}{\text{hr.}}$$

Part (A)



The volume flow rate is the same as the initial condition $756 \text{ ft}^3/\text{hr.}$

Mass Flow Rate:

$$\dot{m} \frac{\text{lb.}}{\text{hr.}} = \frac{\dot{V} \frac{\text{ft}^3}{\text{hr.}}}{v_1 \frac{\text{ft}^3}{\text{lb.}}}$$

$$\dot{m} = \frac{756}{1.184} = 639 \frac{\text{lb.}}{\text{hr.}}$$

(ACTUALLY LESS DUE TO HIGHER PRESSURE RATIO AND LOWER η_{vol})

BUT C_L IS UNKNOWN
COULD ESTIMATE USING ASSUM C_L

Cooling Capacity:

$$\dot{Q}_{Btu/hr} = \dot{m} \frac{\text{lb.}}{\text{hr.}} (h_1 - h_4) \frac{\text{Btu}}{\text{lb.}}$$

$$= 639 (82.40 - 28.74)$$

$$= 639 \cdot 53.66$$

$$= 34,300 \frac{\text{Btu}}{\text{hr.}}$$

Part (B)

Indicating rate of entropy increase at initial conditions:

$$S_1 = 0.1732 \frac{\text{Btu}}{\text{lb.}^\circ\text{R}} \quad S_2 = 0.1776 \frac{\text{Btu}}{\text{lb.}^\circ\text{R}}$$

$$\Delta S = 0.0044 \frac{\text{Btu}}{\text{lb.}^\circ\text{R}}$$

$$\text{rate of entropy increase} = \frac{\Delta S}{\Delta P} = \frac{0.0044}{75} = 5.88 \times 10^{-5} \frac{\text{Btu}}{\text{lb.}^\circ\text{R psi}}$$

final entropy for part (a)

$$\begin{aligned}\Delta S &= \text{Rate} \times \Delta P \\ &= 5.98 \times 10^{-5} \times 90 \\ &= 5.3 \times 10^{-3} \\ &= 0.0053 \frac{\text{Btu}}{\text{lb}^\circ\text{R}}\end{aligned}$$

$$\begin{aligned}S_2 &= S_1 + \Delta S \\ &= 0.1741 + 0.0053 \\ &= 0.1794 \frac{\text{Btu}}{\text{lb}^\circ\text{R}}\end{aligned}$$

C.O.P. for initial conditions:

$$\text{C.O.P.} = \frac{84.60 - 28.74}{94.24 - 84.60} = \frac{55.86}{9.64} = 5.78$$

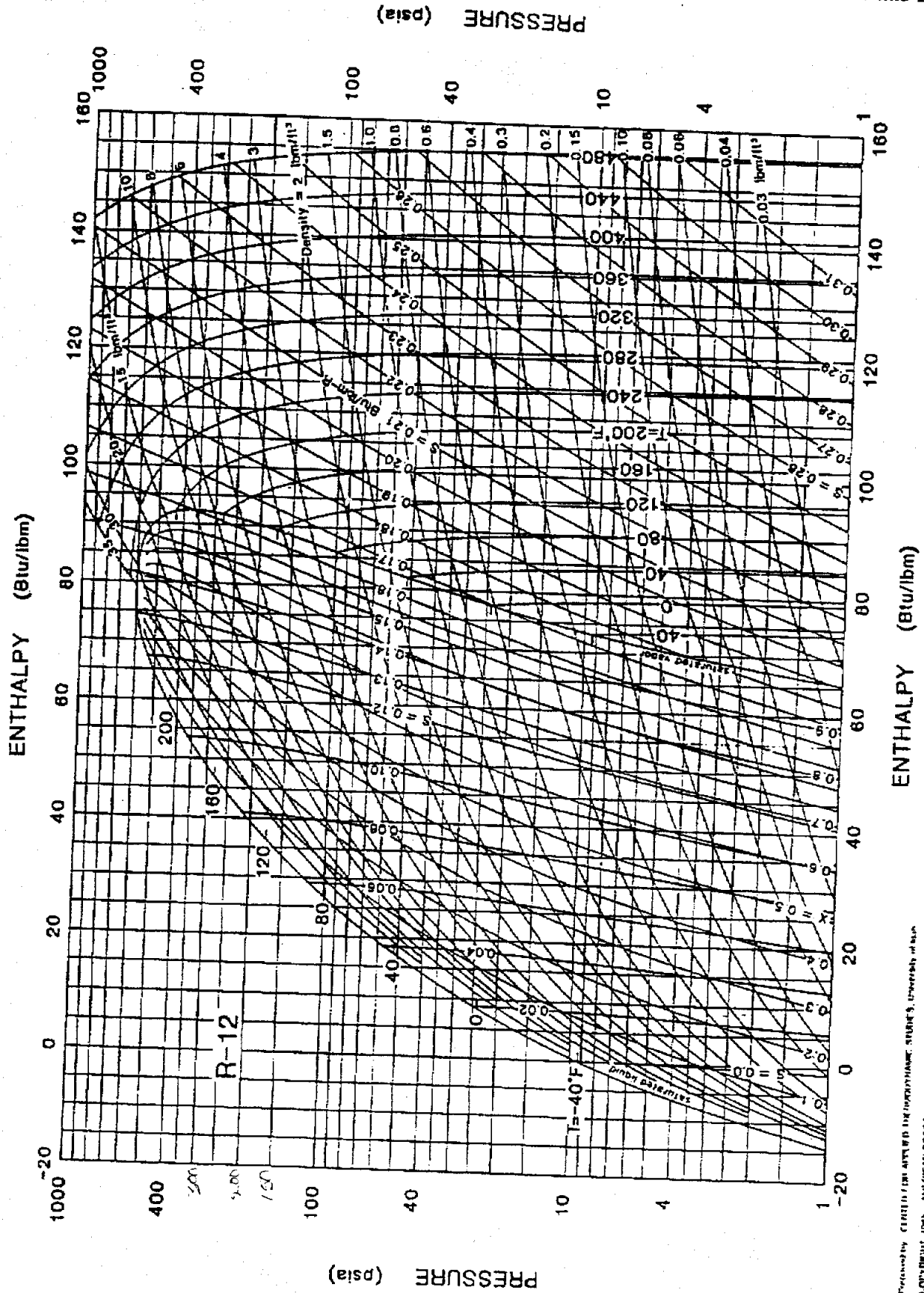
C.O.P. for part (A):

$$\text{C.O.P.} = \frac{82.40 - 28.74}{95.97 - 82.40} = \frac{53.66}{13.57} = 3.96$$

OR

BASED ON COMPR. EFF ~~(TEMPERATURE)~~

$$\eta_{T_1} = \eta_{T_2}$$



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Fig. 2 Pressure-Enthalpy Diagram for Refrigerant 12

ADDENDUM TO PROBLEM NCEE APRIL 1971

$$K = \frac{C_p}{C_r} = \frac{.1516}{.1305} = 1.16$$

ASSUMING $C_L = .05$

$$\epsilon \quad \eta_{V1} = 1 + C_L - C_L \left(\frac{P_2}{P_1} \right)^{1/K} = 1.05 - .05 \left(\frac{125}{35} \right)^{1/1.16} = .94$$

$$\eta_{V2} = 1.05 - .05 \left(\frac{125}{35} \right)^{1/1.16} = .90$$

ADJUSTING FOR CHANGE IN η_V :

$$\therefore \text{NEW COOLING CAPACITY} = 34300 \times \frac{.90}{.94} = 32400 \text{ BTU/HK}$$

USING ADIABATIC COMPRESSOR EFFICIENCY INSTEAD OF ENTROPY CHANGE RATE:

$$\eta_{C1} = \frac{92.03 - 84.6}{94.24 - 84.6} = .771$$

ASSUME $\eta_C = \text{CONST}$ I.E. $\eta_{C1} = \eta_{C2}$

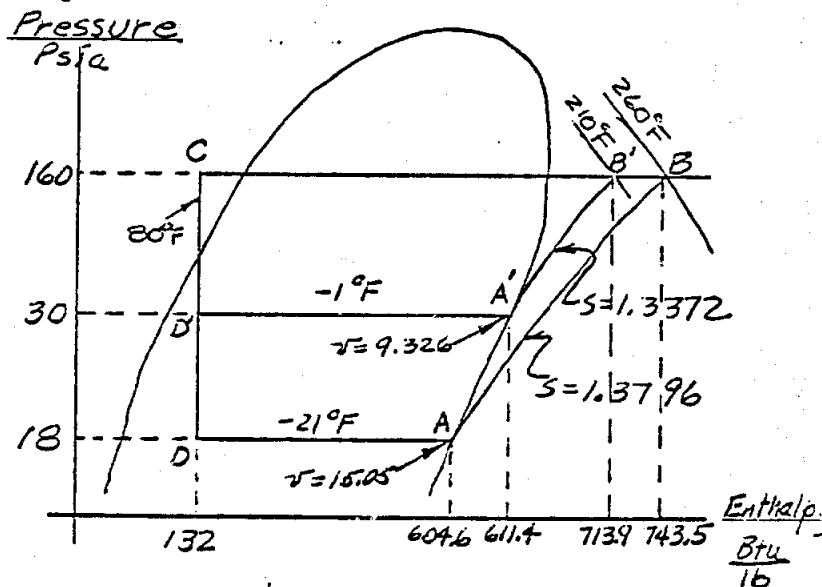
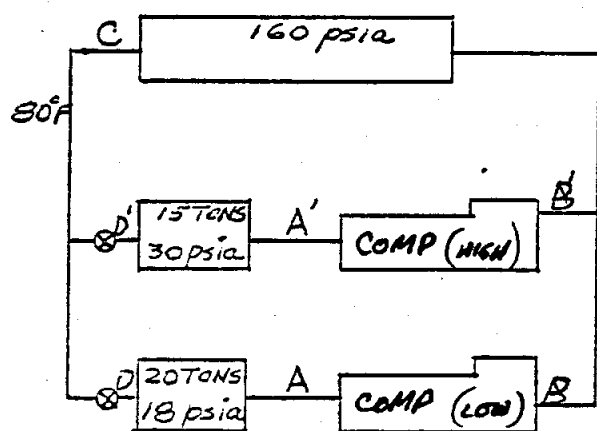
$$\therefore h_2 = 82.4 + \frac{92.57 - 82.4}{.771} = 95.6 \text{ (9\%)}$$

REFRIGERATION PROBLEM FROM APRIL, 1969 P. E. EXAM

A packing plant using ammonia as the refrigerant requires 15 tons of refrigeration at an evaporator pressure of 30 psia, and 20 tons at 18 psia. Each evaporator, by means of its own compressor, discharges to a common condenser operating at a pressure of 160 psia. The ammonia leaving the condenser is subcooled to 80 F, after which it passes through separate expansion valves to the evaporators.

Using an ideal vapor-compression cycle, and assuming saturated vapor at the inlet of each compressor, find:

- The overall horsepower required per ton of refrigeration
- The displacement of each compressor.



Low Pressure System

Evaporator

$$\dot{Q}_{\frac{\text{Btu}}{\text{min}}} = \dot{m}_{\frac{\text{lb}}{\text{min}}} (h_A - h_D)_{\frac{\text{Btu}}{\text{lb}}}$$

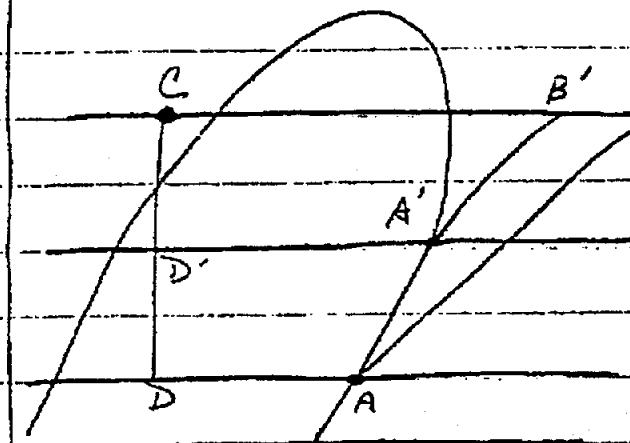
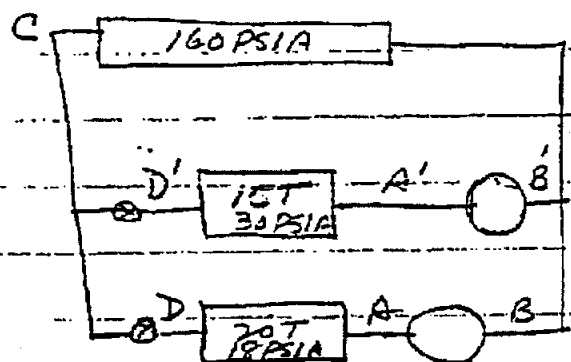
$$\dot{m} = \frac{\dot{Q}}{h_A - h_D} = \frac{20 \times 200}{604.6 - 132} = \frac{4000}{472.6} = 8.45 \frac{\text{lb}}{\text{min}}$$

Compressor

$$\dot{W}_x \frac{\text{Btu}}{\text{min}} = \dot{m}_{\frac{\text{lb}}{\text{min}}} (h_A - h_B)_{\frac{\text{Btu}}{\text{lb}}}$$

$$\dot{W}_x \frac{\text{Btu}}{\text{min}} = 8.45 (604.6 - 743.5) = 8.45 \times (-138.9) = -1171 \frac{\text{Btu}}{\text{min}}$$

$$\text{H.P.} = 1171 \frac{\text{Btu}}{\text{min}} \times 2.356 \times 10^{-2} \frac{\text{HP}}{\frac{\text{Btu}}{\text{min}}} = 27.6 \text{ HP}$$



POINT	P	T	S	h	v
A	18	-21	2.56	200	14.9
A'	30	-1	2.52	205	9.23
B	160	260	2.56	343	
B'	160	213	2.52	305	
C	160	80		-275	
D	18	-21		-275	
D'	30	-1		-275	

LOW PRESS. SYSTEM

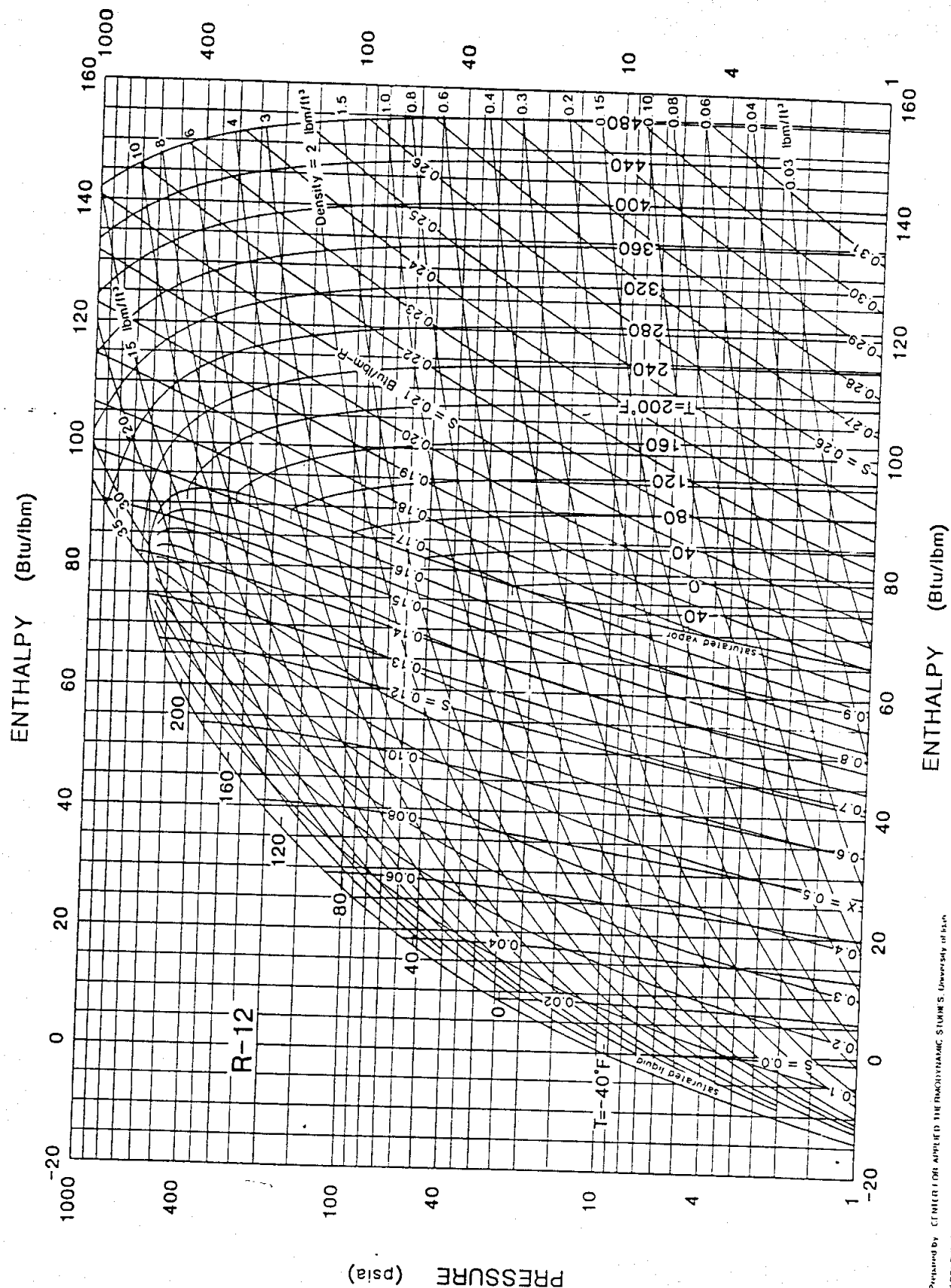
$$Q_{EVAP} = \dot{m}(h_A - h_D)$$

$$\dot{m} = \frac{20 \times 200}{200(-275)} = 8.4 \text{ LB/MIN}$$

COMPRESSOR $\dot{W} = \dot{m}(h_A - h_C)$

$$= 8.4 (343 - 200) = 1204 \text{ BTU/MIN}$$

$$= 1204 \times 0.2356 = 28.4 \text{ HP}$$



Prepared by CENTER FOR APPLIED THERMODYNAMIC STUDIES, University of Illinois
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Fig. 2 Pressure-Enthalpy Diagram, R-12

HIGH PRESS. SYSTEM

$$\text{EVAP} \quad \dot{m} = \frac{15 \times 200}{205 - (-275)} = 6.3 \text{ LB/MIN}$$

$$\text{COMPR.} \quad \dot{W} = 6.3 \times (205 - 305) = 630 \text{ BTU/MIN} \\ = 14.7 \text{ HP}$$

$$\text{OVERALL HP/TON} = \frac{28.4 + 14.7}{20 + 15} = 1.23$$

COMPRESS. DISPL.

$$\text{LOW} \quad \dot{V} = 8.4 \times 14.9 = 125.2 \text{ CFM}$$

$$\text{HIGH} \quad \dot{V} = 6.3 \times 9.23 = 58.1 \text{ CFM}$$

A refrigeration system is to be designed to produce a cooling load of 20 tons of refrigeration. The working medium is to be Freon F-12. The pressure in the evaporator is 15 psia and the temperature leaving the evaporator is 20°F. The temperature

leaving the compressor is 180°F and the pressure is 150 psia. For the simple cycle, the conditions entering the compressor are the same as those leaving the evaporator. Neglect losses for both heat transfer and pressure drop in the lines and for pressure drop in the condenser and evaporator. The temperature leaving the condenser is 100°F. Calculate the following for the simple cycle:

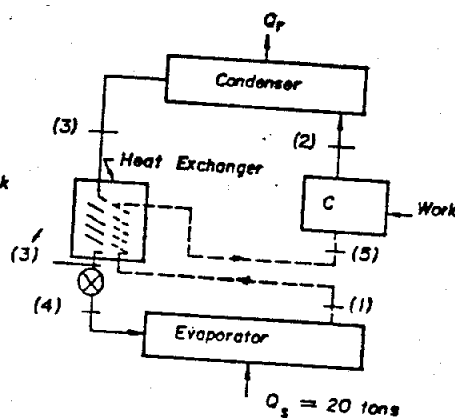
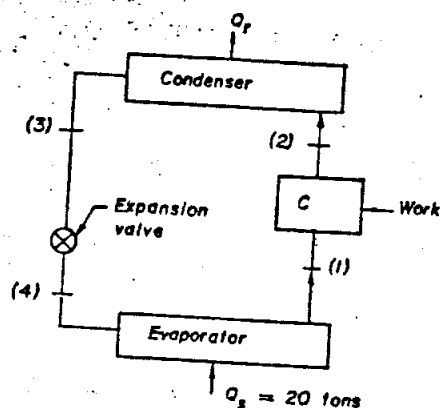
- The required mass rate of Freon flow in lbm/min.
- The required compressor displacement for a volumetric efficiency of 80%. Give answer in cfm.

If a regenerative heat exchanger can be used, as illustrated in the sketch, to increase the temperature entering the compressor to 40°F with no change in conditions leaving the evaporator or condenser, neglect losses and pressure drop and find:

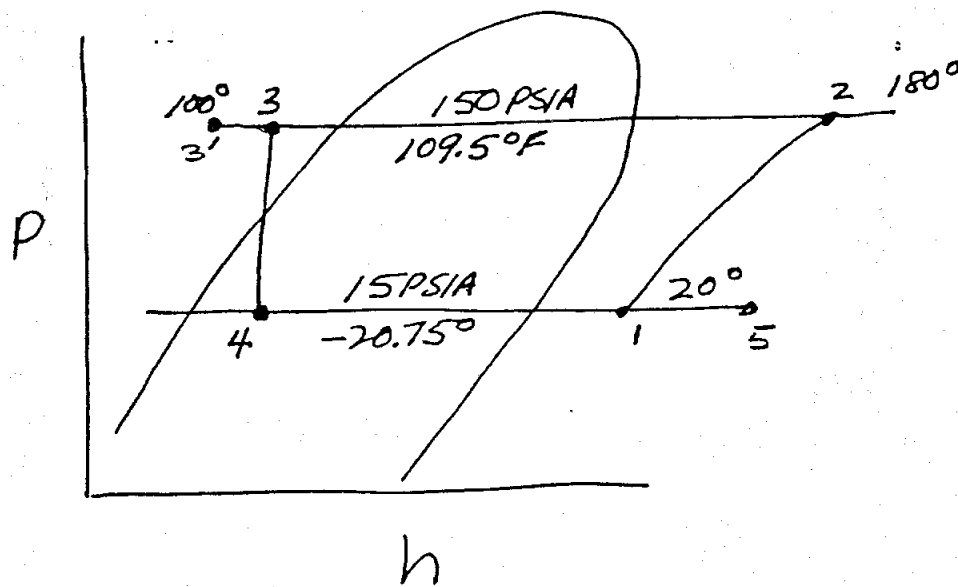
- The new mass rate of Freon flow required in lbm/min.
- The new required compressor displacement at the same 80% volumetric efficiency. Give answer in cfm.

Please use same station numbers as indicated on the sketch.

State reference used for table or chart values.



PROB # 446 NCEE 72-76



$$h_1 = 80.71 \quad S_1 = .1835 \quad v_1 = 2.7494 \text{ ft}^3/\text{lb}$$

$$h_2 = 100.73 \quad S_2 = .1845 \quad v_2 = .332 \text{ ft}^3/\text{lb}$$

$$h_3 = h_4 = 31.22 \text{ (EVALUATED @ } 100^\circ \text{ SAT.)}$$

$$Q. \quad Q_{\text{REFRIG}} = \dot{m}(h_1 - h_3)$$

$$\dot{m} = \frac{20 \times 200}{80.71 - 31.22} = \frac{4000}{49.49} = 80.8 \text{ LB/MIN}$$

$$b \quad \dot{V}_{REQ} = \dot{m} v = 2.7494 \times 80.8 = 222 \text{ FT}^3/\text{MIN}$$

$$\dot{V}_{ACT} = \frac{222}{\eta_v} = \frac{222}{.8} = 277.8 \text{ CFM}$$

$$\eta_v = \frac{\dot{m}_{ACT}}{\dot{m}_{THEOR}} = \frac{\dot{V}_{ACT}}{\dot{V}_{THEOR}}$$

c AT 40°F & 15 PSIA

$$h_5 = 83.56 \text{ BTU/LB} \quad v_5 = 2.877 \text{ FT}^3/\text{LB}$$

$$\therefore h_{3'} = h_3 - (h_5 - h_1) \text{ ENERGY BAL. ON HEX}$$

$$= 31.22 - (83.56 - 80.71) = 28.37$$

$$\dot{m} = \frac{20 \times 200}{80.71 - 28.37} = 76.4 \text{ LB/MIN}$$

NOTE: h_1 NOT h_5

$$d \quad \dot{V}_{ACT} = \frac{76.4 \times 2.877}{.8} = 274.8 \text{ CFM}$$